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HEAT EXCHANGER DESIGN FOR THERMAL CYCLE FEASIBILITY EVALUATION

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THERMAL CYCLE FEASIBILITY EVALUATION

by

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Heat Exchanger Design for Thermal Cycle Feasibility Evaluation

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ABSTRACT

The general properties of heat exchangers are examined in this study. Mathematical models are developed for the mechanism of heat transfer that take place during

1. Evaporation in two-phase flow
2. Condensation
3. Heating or cooling

From these models, computer programs have been developed for the preliminary designing of the following heat exchanger types:

1. Counter-flow cooler with no change of state
2. Counter-flow heater with no change of state
3. Counter-flow condensor
4. Counter-flow evaporator with two-phase flow
5. Cross-flow heater
6. Cross-flow cooler

These programs are designed to be used in the examination of the size requirments for heat exchangers used in the operation of thermodynamic cycles. The cycles of interest in this study are ones proposed for the production of useful power from sources not conventionally used at present.

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I. INTRODUCTION

In recent years, many proposals have been made for methods to produce power from sources not conventionally used. With the recent increase in the cost of fossil fuel energy sources, it can be expected that many such proposals will be forthcoming in the near future.

One such proposal came from Professor Clarence Zener of Carnegie-Mellon University(1). This proposed cycle would produce power utilizing the temperature gradient in the tropical areas of the oceans. To do this, a working fluid with a high vapour pressure is heated by the water at the surface of the ocean. In the tropics, this is around 25°C. The working fluid, as a vapor, then drives a turbine to produce power. It is then condensed back into a liquid by the deep ocean water at a temperature of 5°C.

Hilbert and James Anderson(2) in 1966 proposed a cycle similar to that of Professor Zener. It relies on the heating of the ocean surface water by the sun as the means for powering the system. In 1966, the cost of such a power source was deemed unrealistic, therefore the project was discontinued. In the light of the current shortage of energy, this proposal might warrent re-examination.

Another such proposal came from a husband and wife team. Aden and Marjorie Meinel, optical physicists, proposed a different way of using the solar energy to produce unsable power(3).

Their proposed cycle would use solar energy collectors to heat a liquid metal. This liquid metal would heat the working fluid which drives a turbine to produce power.

The use of geothermal energy, heat from the earth's interior, as a power source has also been suggested(4). This type of heat source would be used in a manner similar to the warm surface water in the Zener proposal.

All of these proposed cycles require similar machinery. The heat source is used to heat or evaporate the working fluid. For this, a heat exchanger is needed. The working fluid then drives a turbine and is cooled or condensed in another heat exchanger. A pump or compressor is used to raise the pressure of the fluid and the cycle begins again.

The feasibility of these and other such proposals must be examined. One phase of this examination should be to obtain an estimation of the size of the equipment necessary for the operation of the cycle. A major item, both in the performance of the cycle and the size of the equipment, is the heat exchangers that are required.

It is proposed that this study be an examination of the required heat exchangers. This will be carried out by the development of a computer program for preliminary design purposes. General properties of heat exchangers will be examined. Mathematical models will be developed for evaporation, condensation, heating and cooling heat exchanger design.

The heat exchangers needed for the cycle proposed by

Professor Zener will be designed in this study. The author feels that this will be useful for two major reasons. First, it will serve as an example of how this study and the computer programs developed can be used. A careful examination of the Zener proposal can also be useful in determining if it is a feasible method of producing power.

II. HEAT EXCHANGER PROPERTIES

2.1 General Properties

2.1.1 Introduction

A heat exchanger is a device used to transfer heat between two fluids that are separated by a wall. It is the form of the separating wall and the flow pattern of the fluids that distinguish between types of exchangers. This study will examine shell and tube heat exchangers and tubular cross flow exchangers. Fins are attached to the walls of some heat exchangers to increase the heat transfer. The use and effectiveness of fins will also be examined.

Cost evaluation is an important problem in the heat exchanger design. A minimum cost, including intitial cost and the cost of pumping the fluids, is generally desired. The alternatives in cost are examined in this study.

2.1.2 Shell and Tube Exchangers

A shell and tube heat exchanger is made up of round tubes mounted in a cylindrical shell. The axis of the tubes are parallel to the shell. Shell and tube exchangers are classified by the direction of flow of the fluids and by the method of reducing thermal stress between the tubes and the shell used (5 (page 18-34)). Selection of the type of exchanger to be used depends on a variety of properties including cost, pressures and temperatures.

Fixed-tube-sheet exchanger is the simplest to fabricate and has the least cost. In this type of heat exchanger the shell and

and tubes are rigidly held. Because of this method of connection the thermal stresses are the highest of any type of shell and tube exchanger. Another disadvantage of fixed-tube-sheet exchangers is the inability to clean the shell side of the exchanger by mechanical means.

To eliminate the high stresses of a fixed-tube-sheet exchanger, an exchanger with an internal floating head can be used. With this type of exchanger the tubes can be removed for cleaning. The cost of an internal floating head exchanger is high and there is much more of a leaking problem than with a fixed exchanger. Fot the same job, the two types of exchangers would have the same size requirements and the design method is the same.

The most commonly used shell and tube exchanger is the fixed-tube-sheet type(6). For this study it will be examined.

A factor that has a large effect on the design of a heat exchanger is the choice of fluid to be passed through the tubes and that to be passed on the shell side. A number of factors go into the making of this decision: (5(Page 18-35)).

1. Cleanability. The shell is difficult to clean and therefore should have the cleanest fluid.

2. Pressure. It is much easier and cheaper to made tubes to withstand high pressure than the shells. High pressure fluids should pass in the tube.

3. Temperature. High temperature fluids cause high stress. This stress can be more easily allowed for in the tubes. High temperature fluids should pass through the tubes.

4. Quantity. The best design can be obtained when the fluid with the smallest flow rate is placed in the shell.

A counter heat flow exchanger has the flow of the two fluids in opposite directions. Figure 2.1-1. In most exchangers the flow is not purely counter flow. In the entrance and exit regions of the exchanger, some cross flow exists. If the exchanger is more than twice as long as it is wide, this cross flow region is small and can be disgarded.

Parallel flow exchangers have the flow of both fluids in the same directions. See Figure 2.1-1. As with the counter flow exchanger, the end regions have some cross flow which is disregarded. Parallel flow exchangers require more heat transfer surface area, for the same temperature rise, than counter flow. See Figure 2.1-2. Parallel flow also has the disadvantage, compared to counterflow,,that the fluid being heated can only have a temperature rise equal to fifty percent of the hot fluid temperature at inlet. Counter flow exchangers can, in the limiting case, have the fluid being heated reach the hot fluid inlet temperature(7). For these reasons parallel flow exchangers will not be further examined in this study.

2.1.3 Cross Flow Exchangers

In a single pass cross flow heat exchanger, the two fluids move perpendicular to each other. Figure 2.1-1. Cross flow generally requires more heat transfer area than counter flow heat exchangers for the same temperature rise. Figure 2.1-1. Cross flow units are used in areas of special application as the exchanger can be made compact. These exchangers are most widely used in gas to liquid exchangers with finned surfaces(8). (More discussion of fins appears in section 2.1.4).

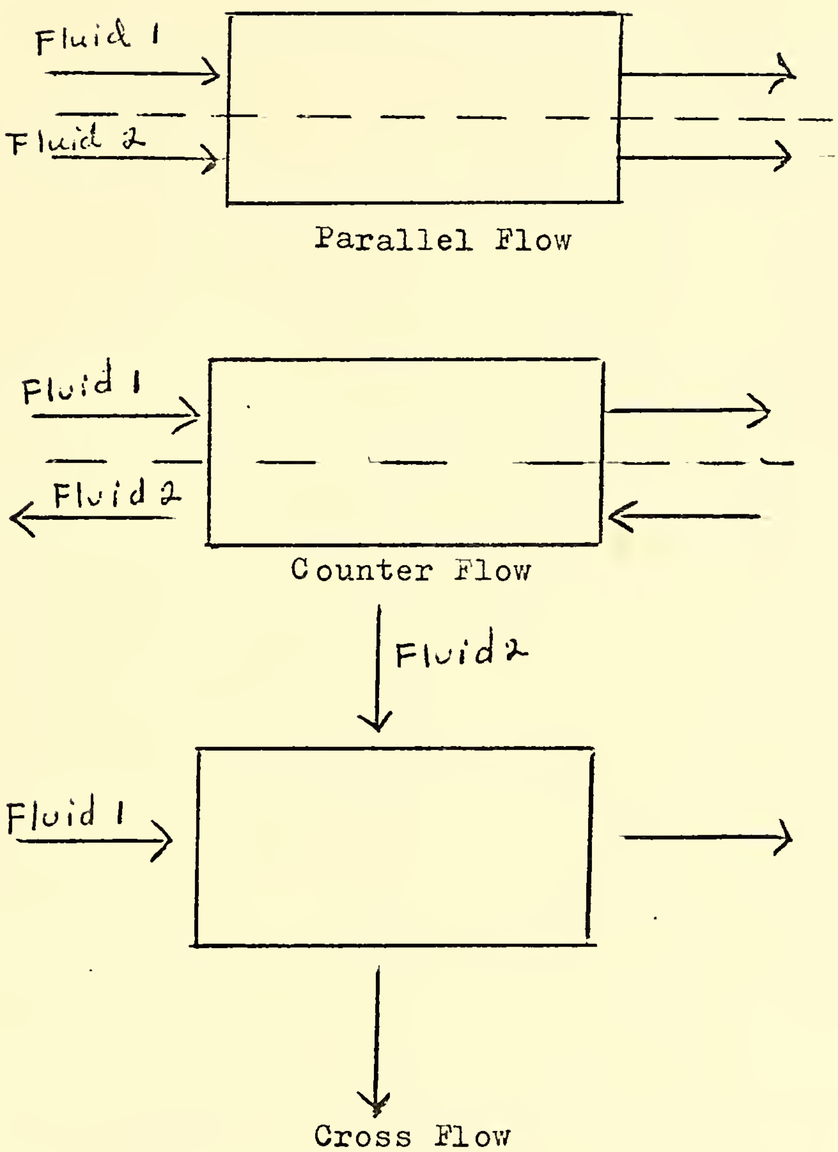
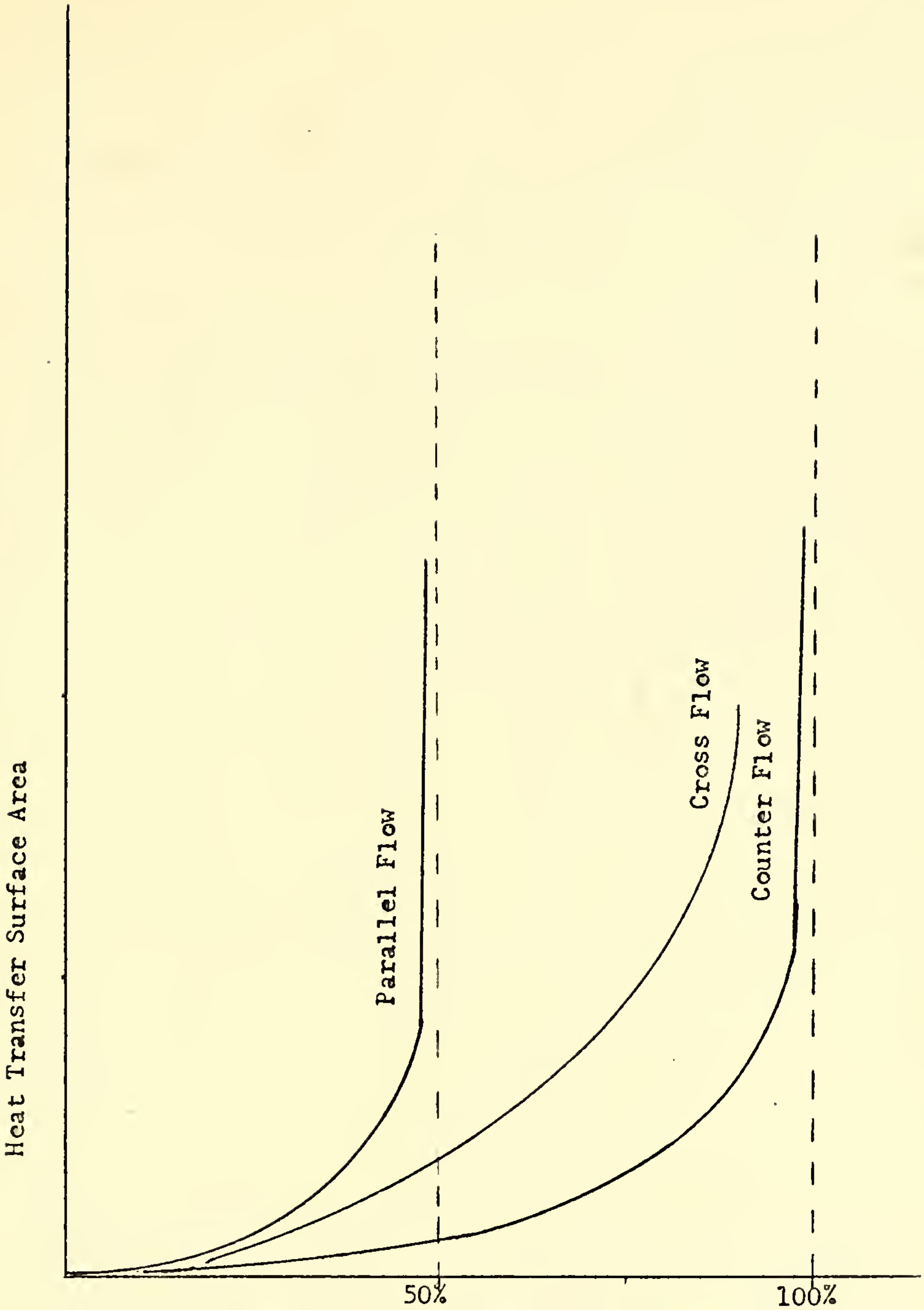


Figure 2.1-1
Flow Patterns of Heat Exchangers



Cold Fluid Temperature Rise as a Percentage of
Hot Fluid Temperature Loss

Figure 2.1-2

Comparison of Required Heat Transfer Area

The cross flow heat exchangers examined in this study will be tubular exchangers. The heating or cooling fluid is passed through a tube and remains unmixed. The working fluid flows across the tubes and is allowed to mix.

2.1.4 Finned Surfaces

Fins are extensions of the tubes that increase the heat transfer surface. They are most effectively used when the heat transfer is between a liquid and a gas. A rough rule-of-thumb for the heat transfer surface area of the fin is that the surface area for the gas and liquid should be inversely proportional to the ratio of respective heat transfer coefficient(7(page 182)). This rule must be slightly modified because the efficiency of a fin decreases with an increase in area. If, for example, an air-water heat exchanger was desired, the area of the air side should be 10 to 20 times as large as the water side. This is caused by the heat transfer coefficient of water being about 500 Btu/ hr ft ^{o}F to 20 Btu/ hr ft ^{o}F for air. To make up this need for different areas, fins would be used on the air side.

The spacing of fins in a heat exchanger is generally determined by manufacturing limits. The height is limited by the required flow area. A spacing of from 6 to 15 fins per inch is the normal range(9).

In the heat exchangers examined in this study, fins are not used. Cost is an important factor. The fabrication costs of fins is much higher than for non-finned tubes. If the fin

is not either integrally or metallurgucally bonded to the tube, the fin efficiency may be adversely affected(11). This requires the fins to be soldered or welded to the tubes. This is an expensive fabrication process. Because fins decrease the flow area, they increase the required pumping power for the fluids, increasing the operational costs.

The nature of the fluids used in this study also discourage the use of fins. In the proposed cycles, the heat transfer is from a fluid in one phase to a fluid of the same flow or to a two phase flow. Fin performance in both cases is less than ideal.

If a detailed design of heat exchangers was going to be carried out, the use of fins should be included. As this study is a preliminary design to determine feasibility, examination of fins is not needed.

2.1.5. Cost of Construction

The estimation of cost of a heat exchanger is a difficult task and is often only a very rough estimate. Many reasons for this exist. Companies that make heat exchangers and components generally consider cost of both materials and fabrication highly proprietry. They are, therefore, quite reluctent to release this data. What available cost data there is, is a number of years old by the time it is published. With the current rate of inflation, data as little as six months old is inaccurate.

Cost of a heat exchanger is sensitive to special requirements. A small matter such as rigorous quality control can

easily double the cost of an exchanger(11). The use of special materials, unusual size materials, or non-standard shapes can greatly change the cost of an exchanger. All of these factors are difficult to include in a cost estimation.

In this study a rough cost is examined. The values are from 1960(12), and are therefore quite inaccurate. They do serve to give a method of comparison between different shapes. If two exchangers are designed to do the same job, examining their comparitive costs, the best can be selected, regardless of the absolute accuracy of the numbers themselves.

Two cost factors are examined; the cost of the material making up the shell, and the tubes, and the cost of fabrication. The fabrication costs include header joint cost (welding), cost of drilling of the header sheet, cost of cutting tubes to size and positioning of the tubes. The cost does not include inspection, sales costs, and shop overhead(12). The material costs are determined on a base size heat exchanger. Corrections are made to adjust the base size to the particular heat exchanger designed. The two correction factors used are the Tube Material Factor and the Tube Length Factor. The tube material correction takes into account the higher costs associated with the use of special materials. The correction for tube length decreases as the length increases. This accounts for the economy of scale.

$$\text{Cost} = CW + CTB$$

where: $CW = CWV \times N$

and $CWV = 3.0$ if Dout is less than ½ in.

$CWV = 12/5 \times (\text{Dout} - .5/12) + 3.0$ if Dout is greater than ½ in.

$$CTB = 15 - \left[\frac{(\text{AREA} - 100)^{\frac{1}{2}}}{5}\right] \times TLF \times \text{AREA} \times TMF$$

and

$$TLF\ 1.3 - \frac{(\text{LENGTH} - 8)^{\frac{1}{2}}}{10}$$

NOMENCLATURE

AREA	Heat transfer surface area
CTB	Cost of materials to build heat exchanger
CWV	Cost of fabrication per tube
CW	Cost of fabrication of heat exchanger
Dout	Outer diameter of tubes
h	Heat transfer coefficient
k	Thermal conductivity of material
L	Length
P	Wetted parameter
q	Rate of heat transfer
T	Temperature
ΔT	Change in temperature
TLF	Tube length factor
TMF	Tube material factor
S	Cross sectional area

Subscripts:

f	With fin
o	Without fin

2.2 EVAPORATOR PROPERTIES

2.2.1 Introduction

Boiling heat transfer is the mode of heat transfer that occurs when a liquid changes phase to a vapor. There are two regimes of boiling, film boiling and nucleate boiling. The form that boiling will take in a given circumstance depends on the degree of superheat of the liquid. Figure 2.2-1 shows typical pool boiling data(10). Pool boiling occurs when the heating surface is submerged beneath the surface of the liquid. In this study the region of interest is where Tw - Tsat is between 5 and 15 degrees, in the nucleate boiling region.

Nucleation is the formation, growth and motion of bubbles. For nucleation to occur in a liquid, that liquid must be superheated, its temperature raised above the saturation temperature. The nuclei is formed on a foreign object in the liquid. This object is usually a cavity on the heating wall but it can be suspended foreign matter with a non-wetted surface(19, page 6). The bubble grows by heat conduction to the liquid-vapor interface. The size it will become is based on a balance between buoyant and surface tension forces.

When boiling occurs in a confined region with force convection present, two-phase flow results. It is this type of boiling heat transfer that is discussed in this study. A mathematical model for two-phase evaporation is developed.

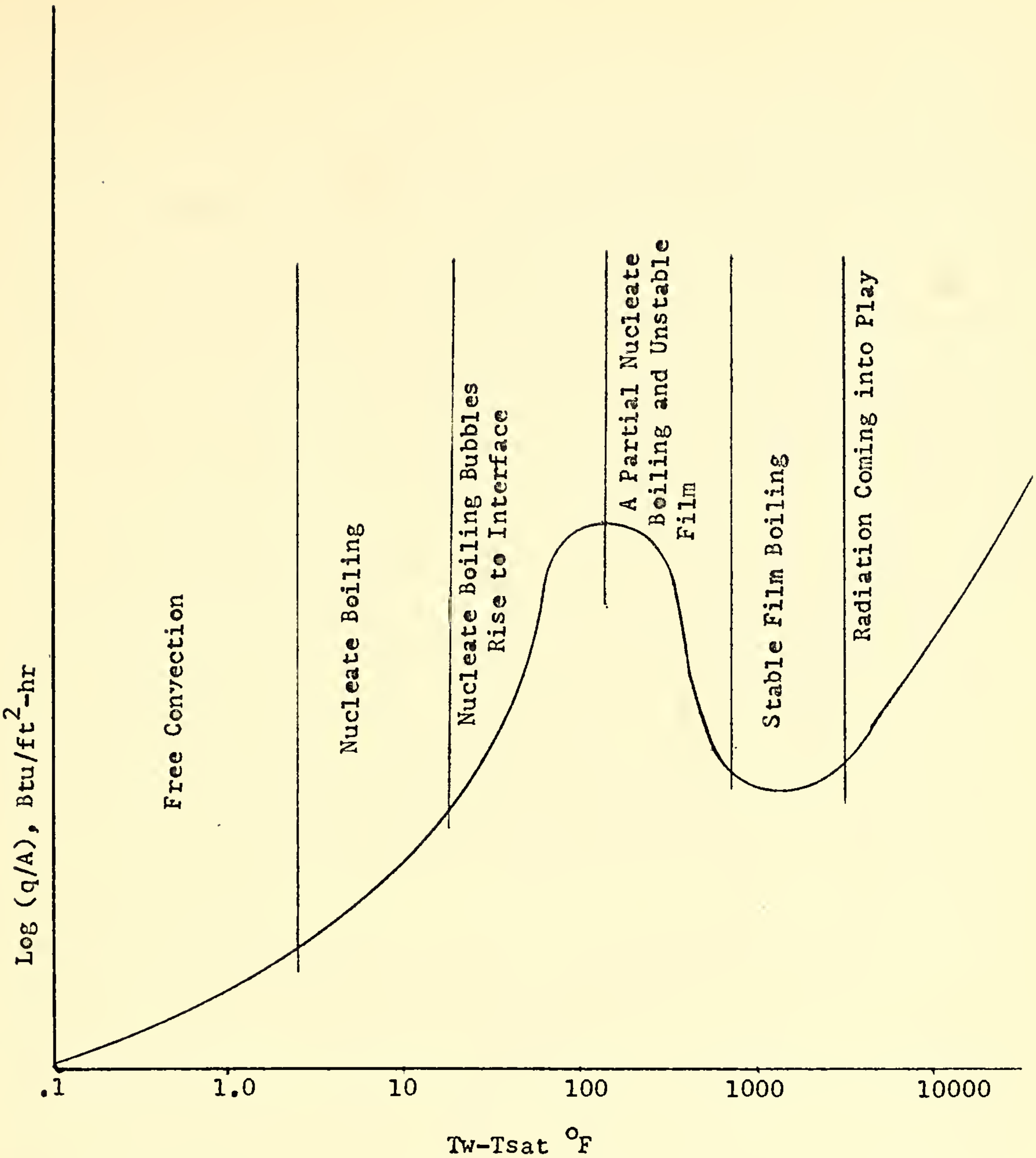


Figure 2.2-1

Pool Boiling Data

2.2.2. Two-Phase Evaporation

Two-phase flow is a pattern that contains both liquid and vapor. It occurs when a flow is subject to forced convection in a confined space, a tube, annuli, parallel plates. The objective of the designer of a two-phase evaporator is to determine the heat transfer and the pressure drop characteristics of this flow. From this data the size of the required heat exchanger can be found.

The determination of the desired characteristics can be a complex task as these two properties are coupled thermodynamically(19, page 47). A change in the heat transfer changes the quality of the flow and the flow pattern. This changes the pressure drop. At the same time a pressure change alters the quality of the flow and therefore effects the heat transfer characteristics.

The flow regimes change as heat is added(20, page 14). Figure 2.2-2. The flow enters as a subcooled liquid. Boiling will take place at the walls even though the liquid as a whole is below the saturation temperature. This forms a bubbly flow. At moderate velocities, greater than 3 feet per second, the bubbles will be fairly evenly distributed. When the liquid reaches the saturation point, bulk boiling will occur. As the quality increases the flow becomes annular with a thin layer of liquid on the walls and a vapor core. In this vapor core will be liquid droplets that will decrease in amount and size as the quality of the flow increases. At some point the heated surface will become

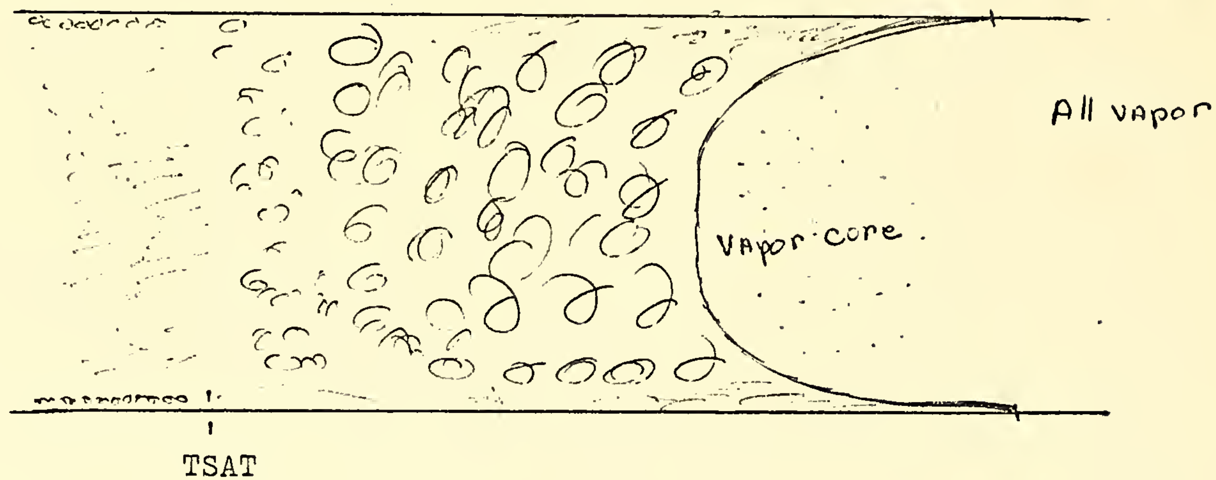


Figure 2.2-2

Two-Phase Flow

dry. This is called the burnout point. From here on the heat transfer mechanism changes and must be dealt with in a separate manner(5, page 13-35).

With the complex flows and the coupling of the heat transfer and pressure drop properties, many of the solution techniques depend on a combination of analytical and experimental studies. A basis for any solution for a two-phase problem is the assumption that the flow is fully developed(19, page 48).

2.2.3 Mathematical Model

Two models can be used for two-phase flow(20, page 24). The 'homogenous' model considers the two phase flow as a single phase with mean fluid properties. This model is also known as the 'friction factor' model. The 'separated flow' model considers the phases to be segregated into two streams, one liquid and one of vapor. If the velocity of each of the streams is considered to be equal, the 'separated flow' model becomes the 'homogenous' model. As one would expect, the 'homogenous' model is valid for the flow of low quality with the separated flow more applicable for the higher quality annular flow. For the mass flow expected in this study, G less than 500lb_m/ft^2sec, the separated flow model gives better agreement with experimental data(20, page 47). For this mathematical model the 'separated flow' model will be used.

The following assumptions are made for the 'separated flow' model: (21)

(overleaf)

1. The vapor velocity and liquid velocity are constant but are not necessarily equal.

2. There is thermodynamic equilibrium between the phases.

3. Empirical correlations can be used to relate the two phase friction multiplier ϕ^2 and the void fraction α to the independent variables of the flow.

It is this third assumption that allows one to separate the calculation of the heat transfer and the pressure drop. The coupling effect of the properties is taken into account by the empirical correlations.

To find the heat transfer in two-phase flow, the standard heat transfer equation $q = A\,U\Delta T$ is used.

Where: $$q = w\,h_{fg}$$

Because there is a great deal of variation in the value of U over the range from a quality, x, of 0 for pure liquid to a quality of 1 for pure vapor, the heat exchanger is broken into segments of equal Δx.

To find U: $$\frac{1}{U} = \frac{1}{h_h} + \frac{1}{h_w} + \frac{Dout - Din}{k_{mat}}$$

The last term is generally disregarded as $\dfrac{k_{mat}}{Dout - Din}$ is much less than h_w and h_h.

To find h_h, the local heat transfer of the heating fluid, the following relationships are used: (22)

For Re < 2100

$$h_h = \frac{1.86\, k_h}{\mu_h^{.5}} \times Pr_h^{.5} \times \frac{G_h^{.5}}{D^{.5}}$$

For $2100 < Re < 10000$

$$h_h = \frac{.116\ k_h\ Pr_h^{.33}}{D} (Re_h^{.67} - 125)$$

For $Re > 10000$

$$h_h = \frac{.023\ k_h\ Pr_h^{.4}\ G_h^{.8}}{D^{.2}\ \mu_h^{.8}}$$

These relationships are empirical equations based on results of many experiments. Other relationships to find h can be used. (10, page 192).

To calculate h_w, the local heat transfer coefficient for evaporation, with two-phase flow is a bit more difficult. Chen(23) postulated that there are two mechanisms which take part in the heat transfer process for boiling in two-phase flow. One is the macroconvective mechanism that is associated with bubble nucleation and growth as takes place in pool boiling. Chen further postulated that these two mechanisms are additive to their contribution to the total heat transfer. Others have suggested this idea. Rohsenow(24) in 1952 and Bambill(25) in 1962 both concluded that the heat transfer effects of boiling and convective flow when occuring at the same time should be additive.

For the contribution of the macroconvective mechanism, Chen(23) felt that the heat transfer coefficient h_{mac} , should be calculated in a manner similar to forced convection flow.

(equation overleaf).

$$h_{mac} = 0.023 \frac{k}{D} Re^{.8} Pr^{.4}$$

This equation is modified to show the effect of the two-phases.

$$h_{mac} = 0.023 \; Re_l^{.8} \; Pr_l^{.4} \; \frac{k_l}{D} \quad F$$

where:

$$Re_l = \frac{G_w \, D \, (1 - x)}{\mu_l}$$

$$Pr_l = \frac{Cp_l \, \mu_l}{k_l}$$

This leaves the only unknown as F. F is the Reynolds factor and is a function of X_{tt}.

where:

$$F = (Re/Re_l)^{.8}$$

and

$$X_{tt} = \left(\frac{1 - x}{x}\right)^{.9} \left(\frac{\rho_v}{\rho_l}\right)^{.5} \left(\frac{\mu_l}{\mu_v}\right)^{.1}$$

For experimental data a curve of F vs $1/X_{tt}$ is plotted. (Figure 2.2-3) An equation can be empirically derived.

$$F = 0.15\left[\frac{1}{X_{tt}} + \frac{2.85}{X_{tt}^{.475}}\right]$$

The contribution of the microconvective mechanism is derived from the analysis of Foster and Zuber(26) on pool boiling. Foster and Zuber found:

(equation overleaf)

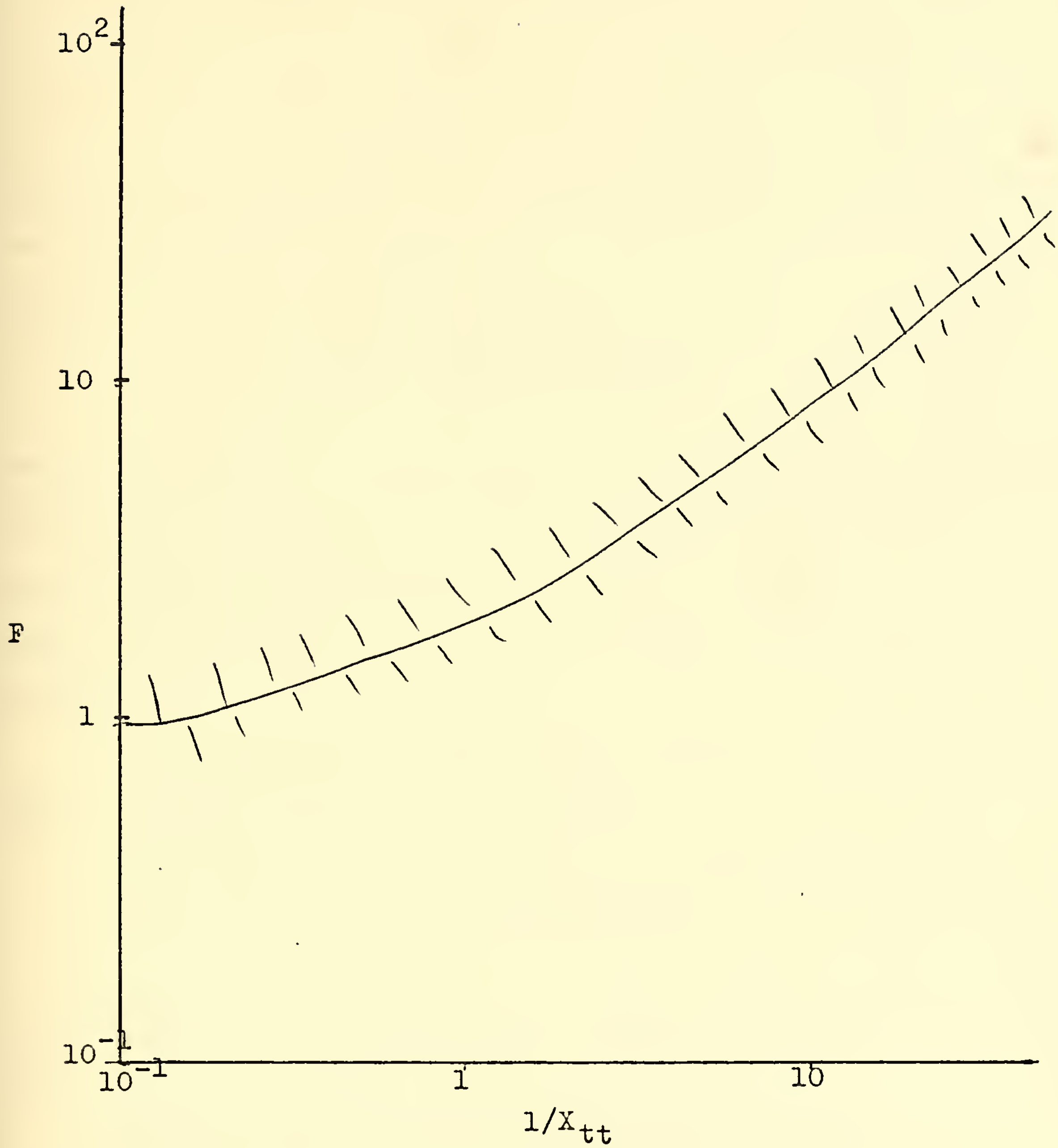


Figure 2.2-3

Reynolds Factor

$$Nu_b = 0.0015\ (Re_b)^{.67}\ Pr_l^{.33}$$

$$= \frac{h_b\ r_b}{k_l}$$

r_b is the radius of the bubble and is derived from bubble buoyancy and surface tension forces.

Where:

$$r_b = \left(\frac{\Delta T\ k_l \rho_l Cp_l \sigma}{\Delta P}\right)^{.5} \left(\frac{\rho_l}{g\ \Delta P}\right)^{.25}$$

Forster and Zuber in their analysis disregarded the fact that the degree of superheat is not constant across the liquid film on the wall material. For convective flow this is important as the temperature gradient across this boundary layer is made much steaper by the flow rate.

Chen(23) then rewrote the Foster and Zuber equation to take this temperature gradient into account.

$$h_{mic} = \frac{0.00122\ k_l^{.79}\ Cp_l^{.45}\ \rho_l^{.49}\ g^{.25}}{\sigma^{5}\ \mu_l^{.29}\ \lambda^{.24}\ \rho_v^{.24}}\ \Delta T^{.24}\ \Delta P^{.75}\ S$$

S is the supression factor and is plotted as a function of $Re = Re_l\ F^{1.25}$.

The total heat transfer coefficient h_w is the sum of h_{mac} and h_{mic}. This correlation appears to give better comparison with experimental data than any other correlations (19, page 126).

The heat transfer coefficient h_w cannot be calculated in the above method for the region that is after the burnout point.

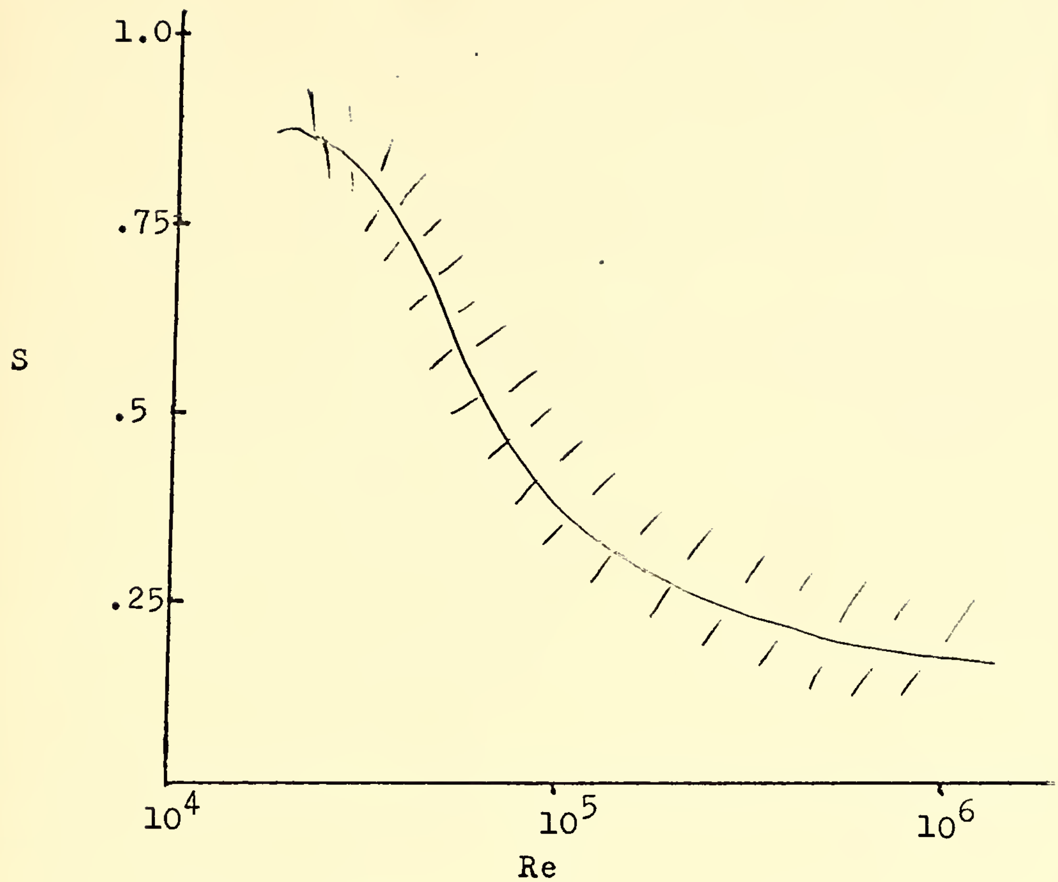


Figure 2.2-4
Supression Factor

For this study it is assumed that burnout will occur at a quality of .95. For the low amount of superheat expected this appears to be a reasonable value. For the segiment of the heat exchanger from quality of .95 to quality of 1, the following should be used for h_w :

$$h_w = \frac{0.023\ k_v\ Pr_v^{.4}\ G_w}{D^{2}\ \mu_v^{8}}$$

Once the value of U is determined, the required heat transfer area can be found for each segment.

$$\text{Area} = \frac{w\ h_{fg}\ \Delta x}{U\ \Delta T}$$

Where:

$$\Delta T = T_{wall} - T_{sat}$$

By assuming a geometry of the heat exchanger, the number of tubes and the diameter of tubes, the length of each segment can be calculated.

$$\Delta z = \text{Area}/\pi D\ n$$

This completes the heat transfer calculations.

To calculate the frictional pressure drop in two-phase flow, the Lockhart-Martinelli-Nelson model(21, 27) will be used. Although this model was derived for adiabatic flow, it is valid for heated pipes(5, page 14 - 9). The heat exchanger is broken into the same segment as for the heat transfer calculations. The pressure drops are then summed for the total.

The relation of two-phase friction pressure drop to single-phase friction pressure drop is:

$$\left(\frac{\Delta P}{\Delta z}\right)_{TPF} = \left(\frac{\Delta P}{\Delta z}\right)_{v} \phi_v^{2}$$

Where ϕ_v^2 can be found empirically as is plotted as a function of X_{tt} . Figure 2.2-5.

$$\phi_v = 1 + 2.85\ (X_{tt}^{\ 0.52\ 2})$$

Then

$$\left(\frac{\Delta P}{\Delta z}\right)_{TPF} = \left(\frac{\Delta P}{\Delta z}\right)_{1} \phi_1^{2}$$

Where ϕ_1^2 can be found empirically

$$\phi_1 = \frac{\phi v}{X_{tt}}$$

$$\left(\frac{\Delta P}{\Delta z}\right)_{TPF} = \phi_v^{2} \times {}^{1.8} \left(\frac{\Delta P}{\Delta z}\right)_{1.0}$$

Where:

$$\left(\frac{\Delta P}{\Delta z}\right)_{LO} = \frac{.09\ G_W^{\ 1.8}\ \mu_v^{\ .2}}{g\ \rho_v\ D^{1.2}} \quad \text{if } Re > 2500$$

And

$$\left(\frac{\Delta P}{\Delta z}\right)_{LO} = \frac{128\ G_W\ \mu_v}{g\ \rho_v\ D^{2}} \quad \text{if } Re < 2500$$

By substitution, $\left(\frac{\Delta P}{\Delta z}\right)_{TPF}$ can be determined. ΔP_W is calculated by multiplying $\left(\frac{\Delta P}{\Delta z}\right)_{TPF}$ by the length of the segment Δz. The sum

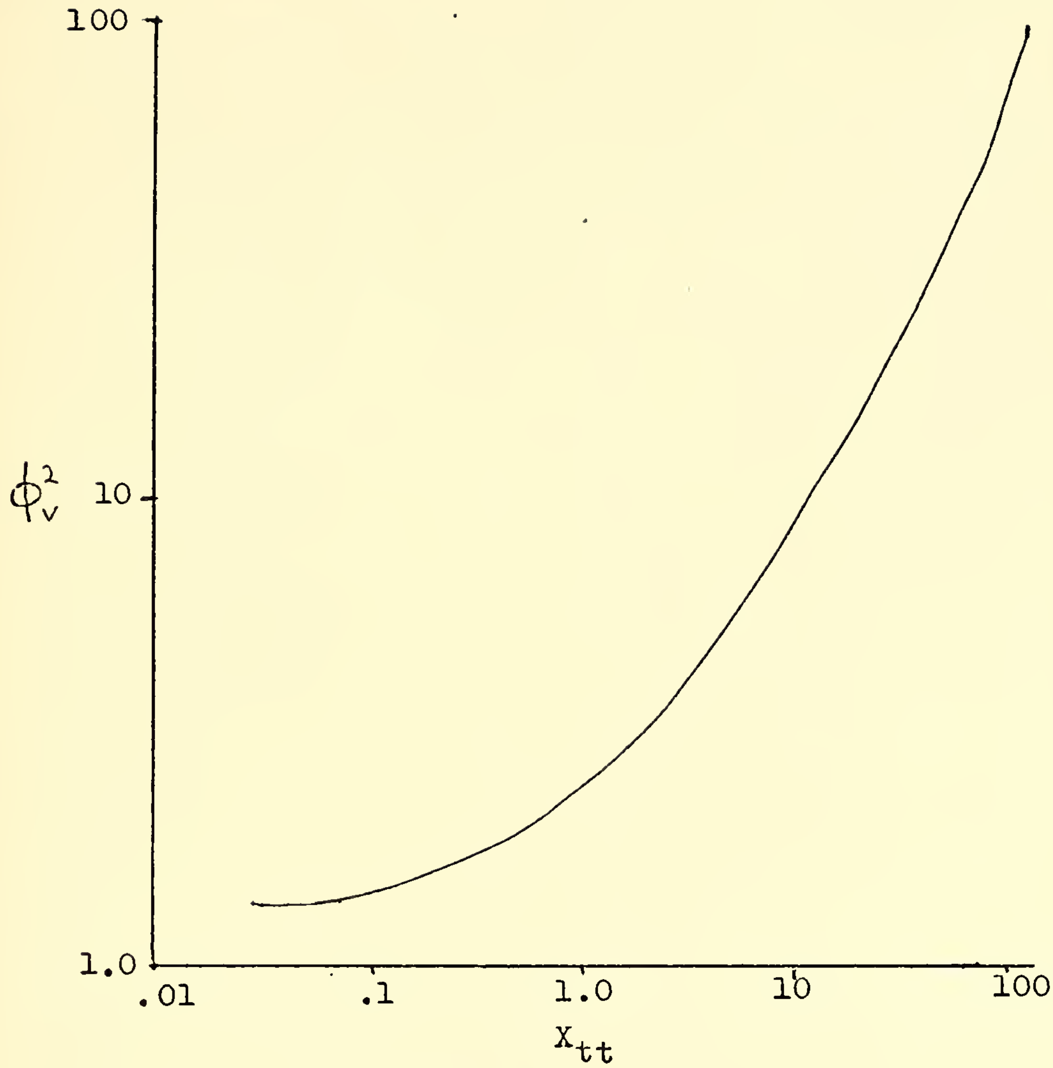


Figure 2.2-5
Two-Phase Frictional Multiplier

of the ΔP_w is the friction pressure drop in the evaporator.

If this pressure drop is too large to be handled by the cycle in which the evaporator is to be placed, the geometry of the evaporator can be changed. Increasing the flow area should decrease the length and the pressure drop.

This concludes the necessary mathematical model for two-phase flow evaporators.

NOMENCLATURE

A	Heat transfer surface area
Cp	Specific heat at constant pressure
D	Diameter
Din	Inter diameter of tubes
Dout	Outer diameter of tubes
F	Reynolds number factor
G	Mass flow velocity
g	Gravitational constant
h	Heat transfer coefficient
h_{fg}	Latent heat of evaporation
k	Thermal conductivity
n	Number of tubes
Nu	Nusselt number
P	Pressure
Pr	Prandtl number
$\frac{\Delta P}{\Delta z}$	Pressure drop per length
ΔP	Change in pressure
q	Rate of heat transfer
r	Radius
Re	Reynolds number
S	Supression factor
T	Temperature
ΔT	Change in temperature

NOMENCLATURE (cont.)

x	Quality
X_{tt}	Martinelli parameter
Δx	Change in quality
w	Mass flow rate
z	Length
λ	Latent heat of vaporization
μ	Viscosity
ρ	Density
σ	Surface tension
ϕ	Two-phase frictional multilplier

Subscripts

b	Bubble
h	Heating fluid
l	liquid
lm	Lograthnic mean
mac	Macroconvective
mat	Material of walls
mic	Microconvective
sat	Saturation
TPF	Two-phase flow
v	Vapor
w	Working fluid

2.3 CONDENSATION PROPERTIES

2.3.1 Introduction

Condensation is the removal of heat from a vapor to convert it into a liquid. This is generally done by using a surface with its temperature, T_w , lower than the saturation temperature, T_{sat} , of the vapor. The vapor will condense into a liquid on the colder surface. The exact nature of this condensation mechanism is not fully established. It is felt that the condensation begins in small cavities in the surface much as bubbles are formed in boiling(5, page 12-2).

There are two forms of condensation - dropwise and film. Dropwise condensation occurs on a non-wetting surface. Droplets form and run down the surface, increasing in size by further condensation and coalescence. New droplets are formed to take their place. The heat transfer for dropwise condensation is from 5 to 50 times that for film condensation(31). It is difficult to maintain dropwise condensation for more than a short period of time without treating the heat transfer surface.

Film condensation occurs when the liquid being formed wets the surface and establishes a stable film. This type of condensation is the prevalent in commercial condensers. As the heat transfer rate is lower with film condensation, condens rs are designed for this form of condensation. If the condensing should be dropwise for part of the operation of the condenser, it will perform better than design. In this study, film condensation

will be examined.

2.3.2 Film Condensation

The first attempt to analyse film condensation was carried out by Nusselt in 1916. When the stable film is formed, vapor condenses on the liquid. The condensation takes place at the liquid-vapor interface because the heat is being transferred through the liquid. At this interface there is a continual interchange of molecules being condensed and molecules evaporating, with the net flow condensing.

In his analysis Nusselt made the following assumptions: (10, page 314).

1. The flow of condensate in the film is laminar.
2. The fluid properties are constant.
3. The condensate temperature distribution is linear.
4. There are no changes of momentum through the film.
5. The vapor is stationary and there is no shear force at the liquid-vapor interface.
6. Heat transfer is by conduction only.

These assumptions are important in the later development of the mathematical model for condensation. The only modification to these assumptions is in the temperature distribution. The actual temperature distribution is slightly curved(10, page 239).

2.3.3 Mathematical Model

In the design of a condenser, the desire is to determine the required heat transfer area. The rate of heat transfer, q, is

found by:

$$q = U\,A\Delta T_{lm}$$

Where: $$\frac{1}{U} = \frac{1}{h_w} + \frac{1}{h_c} + \frac{Dout - Din}{k_{mat}}$$

As was done in boiling heat transfer, K_{mat}/Dout - Din can be disregarded.

$$q = w\,h_{fg}$$

and $$\Delta T_{lm} = \frac{(T_{sat} - T_{c1}) - (T_{sat} - T_{c2})}{\ln\left(\dfrac{T_{sat} - T_{c1}}{T_{sat} - T_{c2}}\right)}$$

Where ΔT_{lm} is the 'logarithic' mean temperature.

To find the required area, all that needs to be calculated is the overall heat transfer coefficient, U. The heat transfer coefficient for the cooling fluid, h_c, is found by the following relations(22).

if $Re_c < 2100$

$$h_c = 1,86(Re_c^{.5}\ Pr_c^{.5}\ \frac{D^{.5}}{L^{.5}})\ \frac{k_c}{D}$$

if $2100 < Re_c < 10000$

$$h_c = .116\ \frac{k_c}{D}\ (Re_c^{.67} - 125)\ Pr_c^{.67}$$

(cont. overleaf)

if $Re_c > 1000$

$$h_c = \frac{0.023\ k_c\ Pr_c^{.4}\ G_c^{.8}}{\mu_c^{.8}\ D^2}$$

To derive the coefficient of heat transfer for condensation, h_w, one begins with the Nusselt analysis. For condensing on a verticle plate,

$$h_w = 0.943 \sqrt[4]{\frac{g\rho_1(\rho_1 - \rho_v)\ k_1^3\ h'_{fg}}{L\mu_1\ \Delta T}}$$

Where $h'_{fg} = h_{fg} + .68\ Cp\ \Delta T$.

The deriviation of this equation is examined in detail in References 5, 31 and 10.

This coefficient must be modified for condensation on a horizontal tube. The relation above is valid with the modification that g sin is used to replace g. The average value of h for the tube is found by integrating h_w from 0 to

$$h_w = 0.728 \sqrt[4]{\frac{g\rho_1(\rho_1 - \rho_v)k_1^3\ h'_{fg}}{Dout\mu_1\ \Delta T}}$$

For a number of tubes in a vertical row, all of the condensate dropping from any tube is assumed to fall on the next lower tube. This will increase the film thickness of lower tubes and require a modification in h_w .

$$h_w = 0.728 \sqrt[4]{\frac{g\rho_1(\rho_1 - \rho_v)\ k_1^3\ h'_{fg}}{n_b\ Dout\ \mu_1\ \Delta T}}$$

n_b is the number of tubes in a vertical row. Observations by Grant and Oswent(34) show that the condensate seldom falls from the upper tube to lower tubes as a continuous sheet. This would tend to lessen the importance of having the n_b term.

Two other effects can change the heat transfer coefficient. These are the presence of noncondensable gas and changes in the surface geometry.

The presence of even a small quantity of non-condensable gas in a condensing vapor has a profound effect on the heat transfer coefficient. The non-condensable gas tends to form at the liquid-vapor interface. This increases the partical pressure of the gas at the interface and increases the resistance to heat transfer. The details of the methods to calculate this effect are given in Reference 31. In this study, the effects will be disregarded. In most commercial condensers, ejectors are used to remove these gases and alleviate the problem.

Changes in the surface geometry of the tubes can greatly increase the heat transfer coefficient with film condensation. One of the most promising of these geometry changes is fluting the tubes. This geometry was first examined by Gregorig(33). In experiments Gregorig obtained an h for condensing of 8000 Btu/hr. ft^2 ^oF, a factor of four above what is obtainable with round tubes. Because of the complex shape of these tubes, it is difficult to analytically evaluate the heat transfer coefficient. Instead, experimental data must be used.

Upon finding h_c and h_w, the overall heat transfer coefficient

can be found and the required heat transfer surface area determined. A heat exchanger geometry, tube diameter and number of tubes is chosen by the designer. The condenser length can be found from this.

$$1 = \text{Area}/\pi n \text{ Dout}$$

The frictional pressure drop of the cooling fluid can be calculated.

$$\text{if } Re_c > 2500$$

$$P_c = \frac{0.046}{Re_c^{\cdot 2}} \frac{4 \; 1 \; G_c^2}{D \; 2g \; \rho_c}$$

Where $$Re_c = \frac{Din \; G_c}{\mu_c}$$

$$\text{if } Re_c < 2500$$

$$P_c = \frac{64}{Re_c} \frac{4 \; 1 \; G_c^2}{Din \; 2g \; \rho_c}$$

If this pressure drop is too large for the desired application, the condenser geometry, tube diameter and number of tubes, can be changed to decrease the pressure loss.

This concludes the necessary mathematical model for condensing on the outside of tubes. Condensing on the inside of tubes is not often carried out as it involves two-phase flow and is difficult to analyse.

NOMENCLATURE

A	Heat transfer surface area
Cp	Specific heat at constant pressure
Din	Inner diameter of tubes
Dout	Outer diameter of tubes
g	Gravitational constant
G	Mass flow rate w/Area
h	Heat transfer coefficient
h_{fg}	Latent heat of condensation
k	Thermal conductivity
L	Length
n	Number of tubes
nb	Number of tubes in vertical bank
Pr	Prandtl number
ΔP	Change in pressure
q	Rate of heat transfer
Re	Reynolds number
T	Temperature
ΔT	Change in temperature
U	Overall heat transfer coefficient
w	Flow rate
μ	Absolute viscosity
ρ	Density

Subscripts

c	Cooling fluid

NOMENCLATURE (cont.)

l	Liquid
lm	Logarithmic mean
mat	Material of the walls
sat	Saturation
v	Vapor
w	Working fluid.

2.4 Cooling and Heating Properties

2.4.1 Introduction

In this study, the evaluation of the properties of cooling a working fluid and heating a working fluid in a heat exchanger are carried out at the same time. Thermodynamically the mechanism of heating and of cooling is the same. Any differences that do exist in the relationships will be discussed in the development of the mathematical model.

As was noted in section 2.1, counter-flow heat exchangers and cross-flow heat exchangers will be examined in this study. The heat transfer mechanisms of both exchanger types are similar but the approach to the designing of each is different.

The job to be done by these heat exchangers will be to transfer a specific amount of heat from one fluid to another. If the exchanger is a heater, the working fluid will be the cooler fluid. For a cooler, the working fluid must be the hotter fluid. The mass flow rate, w, of both fluids in each type of exchanger will be specified as will be the inlet and outlet temperatures. This will be true regardless of the flow pattern of the fluids.

2.4.2 Counter-Flow Heaters and Coolers

The mathematical model for the sizing of counter-flow heaters and coolers is based on the solving of similtaneous equations. This is accomplished by setting groups of known constants equal to a large constant, K1, K2, K3, etc. The heat transfer

area, number of tubes, and the exchanger length will be determined in terms of these constants. This method is described in detail in Reference 7.

To make this model valid for both heating and cooling, the hotter fluid in the exchanger is designated by a subscript 1 and the cooler fluid by a subscript 2. In a heater the working fluid is colder and is designated by the subscript 2, while for a cooler the inverse is true. The mathematical relationships are otherwise identical.

A limiting constraint on the design of a counter-flow exchanger is the pumping power required to overcome the friction pressure loss of the fluid flowing in both the tubes and the shell. In this design method, this constraint is set by establishing an allowable pressure drop, ΔP.

If $Re_1 > 2500$

$$\Delta P_1 = \left(\frac{0.092\ \mu_1^{0.2}}{g\ \rho_1} \right) \frac{G^{1.8}\ L}{Din^{1.2}} \tag{35}$$

$$\Delta P_1 \equiv K1 \frac{G_1^{1.8}\ L}{Din^{1.2}}$$

If $Re_1 < 2500$

$$P_1 = \left(\frac{128\ \mu_1^{.2}}{g\ \rho_1 Re_1^{.8}} \right) \frac{G_1^{1.8}\ L}{Din^{1.2}}$$

$$P_1 \equiv \frac{K1\ G_1^{1.8}\ L}{Din^{1.2}}$$

If $Re_2 > 2500$

$$P_2 = \left(\frac{0.092\ \mu_2^{.2}}{g\ \rho_2}\right) \frac{G_2^{1.8} L}{Deq^{1.2}}$$

$$P_2 \equiv K2 \frac{G_2^{1.8}\ L}{Deq^{1.2}}$$

If $Re_2 < 2500$

$$P_2 = \left(\frac{128\ \mu_2^{.2}}{g\ \rho_2\ Re_2^{.8}}\right) \frac{G_2^{1.8}\ L}{Deq^{1.2}} \equiv K2 \frac{G_2^{1.8}\ L}{Deq^{1.2}}$$

Where: $Deq = (1.271\ (s/Dout)^2 - 1)\ Dout$ (7, page 324)

This is for square pitch exchangers with s the spacing between tube centers.

Where: $Re_1 = Din\ (w_1/\ (\pi\ \frac{Din^2}{4})$

and $Re_2 = Deq\ (w_2/\ (\pi\ \frac{Deq^2}{4})$

Next the heat transfer coefficient, h, is calculated. (30, page 442)

If $Re_1 > 10000$

$$h_1 = \frac{0.023\ k_1\ Pr_1^{.4}}{\mu_1^{.8}} \frac{G_1^{.8}}{Din^{.2}} \equiv K3 \frac{G_1^{.8}}{Din^{.2}}$$

If $2500 < Re_1 < 10000$

$$h_1 = \frac{.116\ k_1\ Pr_1^{.33}}{\mu_1^{.8}\ Re_1^{.13}} \frac{G_1^{.8}}{Din^{.2}} \equiv K3 \frac{G_1^{.8}}{Din^{.2}}$$

If $Re_1 < 2500$

$$h_1 = \frac{1.86\ k_1\ Pr_1^{.5}}{\mu_1^{.8}\ Re_1^{.3}} \frac{G_1^{.8}}{Din^{.2}} \equiv K3 \frac{G_1^{.8}}{Din^{.2}}$$

If $Re_2 > 10000$

$$h_2 = \frac{0.023\ k_2\ Pr_2^{.4}}{\mu_2^{.8}} \quad \frac{G_2^{.8}}{Deq^{.2}} \equiv K4\ \frac{G_2^{.8}}{Deq^{.2}}$$

If $2500 < Re_2 < 10000$

$$h_2 = \frac{.116\ k_2\ \ Pr_2^{.33}}{\mu_2^{.8}\ \ Re_2^{.13}} \quad \frac{G_2^{.8}}{Deq^{.2}} \equiv K4\ \frac{G_2^{.8}}{Deq^{.2}}$$

If $Re_2 < 2500$

$$h_2 = \frac{1.86\ k_2\ \ Pr_2^{.5}}{\mu_2^{.8}\ Re_2^{.3}} \quad \frac{G_2^{.8}}{Deq^{.2}} \equiv K4\ \frac{G_2^{.8}}{Deq^{.2}}$$

Then: $\dfrac{\Delta\ P_1}{\Delta\ P_2}$ (both known) $\dfrac{K1}{K2}\left(\dfrac{G_1}{G_2}\right)^{1.8}\left(\dfrac{Deq}{Din}\right)^{1.2}$

By definition $$\frac{G_1}{G_2} = \frac{w_1/Ax_1}{w_2/Ax_2}$$

Where Ax is the cross sectional area of the flow.

$$Ax_1 = \pi/4\ Din^2\ n$$

$$Ax_2 = \pi/4\ Deq^2\ n = \pi/4\ Deq\ Dout\ n$$

By substitution

$$\frac{G_1}{G_2} = \left(\frac{w_1\ \ \ Dout}{w_2\ \ \ Din}\right)^{.4}\left(\frac{K2\ K5}{K1}\right)^{.33} \equiv K6$$

Then $$\frac{G_1}{G_2} = \frac{w_1\ \ Deq\ Dout}{w_2\ \ Din^2}$$

From solving for $\Delta P_1/\Delta P_2$ and G_1/G_2

$$\frac{h_1}{h_2} = \frac{K3}{K4} \left(\frac{G_1}{G_2}\right)^{.8} \left(\frac{Deq}{Din}\right)^{.2}$$

$$= \frac{K3}{K4} \left(\frac{K2\ K5}{K1}\right)^{.33} \left(\frac{Dout}{Din}\right)^{.2} \left(\frac{w_1}{w_2}\right)^{.2} \equiv K7$$

The overall heat transfer coefficient can be calculated by:

$$\frac{1}{U} = \frac{1}{h_1} + \frac{1}{h_2} + \frac{Dout - Din}{k_{mat}}$$

The last term can be disregarded as it is insignificant compared to the other terms. This equation does not include the effect on U of scale deposits. For a preliminary design it can be disregarded.

$$\frac{1}{U} = \frac{1}{h_1} + \frac{1}{h_2}$$

$$\frac{1}{U} = \frac{1}{K3} \frac{Din^{.2}}{G_1^{.8}} + \frac{1}{K4} \frac{K6\ Deq^{.2}}{G_1^{.8}} + \frac{K8\ Din^{.2}}{G_1^{.8}}$$

Where $$K8 = \frac{1}{K3} \times \frac{K6\ Din^{.2}}{K4\ Deq^{.2}}$$

The heat transfer rate

$$q = A\ U\ \Delta T_{1m}$$

Where $$A = n\ \ Din\ L$$

$$q = n\ \pi Din\ U\ L\ \Delta\ T_{1m} = \frac{\pi}{4} Din^2\ n\ G_1\ Cp_1\ \Delta T_1$$

Where ΔT_1 is the difference between the inlet and outlet temperature of the hotter fluid and ΔT_{1m} is the logarithmetic mean temperature.

Then

$$L = \frac{Cp \quad T_1}{4\Delta \ T_{1m}} \frac{G_1 \ Din}{U} \equiv K9 \frac{G_1 \ Din}{U}$$

$$L = K9 \ K8 \ D_1^{1.2} \ G_1^{1.2}.$$

Then by substitution of

$$G_1^{.2} = \frac{L}{Din^{1.2}} K9 \ K8$$

Into

$$\Delta P_1 = \frac{K1 \ L \ G_1^{.8}}{Din^{1\ 2}}$$

$$G_1 = \left(\frac{P_1}{K1 \ K8 \ K9}\right)^{.5} \equiv K10$$

By definition

$$G_1 = w_1/Ax1$$

Therefore

$$n = \frac{w_1 \quad 4}{G_1 \ \pi \ D_1^2}$$

The value of Din and Dout are selected so L can be calculated.

$$L = K9 \ K8 \ Din^{1.2} \ K10$$

and the heat transfer area can be found.

$$A = Din \ n \ L$$

This concludes the necessary calculations for the heat exchanger design. This design method assumes that:

(1) The specific heat of the fluid, Cp is constant.

(2) The overall heat transfer coefficient is constant.

If these factors are not constant, the design approach is modified. The above procedures are carried out using average values of the fluid properties. A value of n and G_1 is determined. The exchanger is then divided into segments such that Cp and U can be assumed to be constant over the segment. The design method is then repeated for each segment with n and G_1 kept constant. The length is determined for each segment. These lengths can then be added to get the overall length of the heat exchanger.

2.4.3 Cross-Flow Heaters and Coolers

In addition to the information about the job to be done by the heat exchanger noted in section 2.4.1, one must know the condition of the flow through the exchanger to size a cross-flow heater or cooler. (3, page 18 - 81). As stated in section 2.1.3, for this study it is assumed that the working fluid is flowing across the tubes and is mixed. This keeps the working fluid at a uniform temperature as it leaves the exchanger.

In the development of the mathematical model, the equation $q = U\,A\Delta T_m$ is again used with a slight modification -

$$q = U\ A\ F\Delta T_{1mc}:$$

(cont. overleaf).

Where:
$$\Delta T_{lmc} = \frac{(T_{h2} - T_{c1}) - (T_{h1} - T_{c2})}{\ln \dfrac{T_{h2} - T_{c1}}{T_{h1} - T_{c2}}}$$

ΔT_{lmc} is the logarithmetic mean temperature for counterflow arrangement. This is not the integrated mean temperature ΔT_m. F is then defined as:

$$F = \Delta T_m / \Delta T_{lmc} \qquad (30)$$

and is a function of ε_c and Z.

Where: ε_c is called the effectiveness of the heat exchanger.

$$\varepsilon_c = T_{c2} - T_{c1}/T_{h1} - T_{c1} \qquad (30)$$

and
$$Z = \frac{T_{h1} - T_{h2}}{T_{c1} - T_{c2}}$$

F can be found on tables as a function of Z and ε_c. F also varies with the flow pattern, if one or both of the fluids are mixed. These tables can be found in References 5, 10 and 30.

The value of q can be found from $q = w\ Cp_1\ \Delta T_1$. All of these values are known. To find the heat transfer surface area:

$$A = q/U\ F\Delta T_{lmc}$$

U can be calculated by the relations for h_1 and h_2 examined in the section 2.4.2 of this study.

By selecting an exchanger geometry, number of tubes and tube

diameter, the length of the required exchanger can be determined.

$$L = \text{Area}/\pi D_{in}\, n$$

The frictional pressure loss is then calculated. The relationships are examined in detail in section 2.4.2 and need not be rederived. If the pressure loss is too great, the geometry of the exchanger can be modified and the length recalculated.

This design method assumes that U is constant throughout the length of the exchanger. If this is not a valid assumption, the exchanger can be broken into segments. The above design procedure is then used for each segment with the sum of each segiment length the total length.

NOMENCLATURE

A	Heat transfer surface area
Ax	Cross sectional flow area
Cp	Specific heat at constant pressure
Din	Inside diameter of tubes
Dout	Outside diameter of tubes
Deq	Equivalent diameter for shell side
F	Mean temperature difference factor
g	Gravitational constant
G	Mass velocity
h	Heat transfer coefficient
k	Thermal conductivity
K	Constant
L	Length
n	Number of tubes
ΔP	Frictional pressure drop
Pr	Prandtl number
q	Heat transfer rate
Re	Reynolds number
s	Tube spacing
T_c	Temperature of cooler fluid
T_h	Temperature of hotter fluid
ΔT_{lm}	Log mean temperature difference
U	Overall heat transfer coefficient

NOMENCLATURE (cont.)

w	Mass flow rate
Z	Temperature coefficient
ε_c	Effectiveness of heat exchanger
μ	Viscosity of fluid
ρ	Density of fluid

Subscripts

1	Hotter fluid
2	Cooler fluid
mat	Tube material

2.5 Material Selection

2.5.1. Introduction

In the selection of material for the tubes and other heat transfer surfaces in the heat exchanger, it is desirable to have the heat transfer coefficient as high as possible. An increase in the heat transfer at the surface of the material produces a corresponding increase in U, the overall heat transfer coefficient. This will decrease the required heat transfer surface area as the area proportional to the inverse of U.

This augmentation of the heat transfer can be accomplished in many ways. The surface of the tubes can be roughened by a number of methods. The heat transfer surface can be left the same but the flow disturbed. This increases the heat transfer by displacement of the fluid or by setting up a vortex flow. Other methods of augmentation include vibration of the heat transfer surface, fluid vibration, and the use of electrostatic fields. Of these methods of increasing the heat transfer, the best appears to be the roughening of the surface material. The vibration and electrostatic methods require outside power to operate and are therefore less attractive. (5(page 10-2)).

2.5.2. Methods of Enhancement

The first method for enhancing the heat transfer to be seriously considered, was increasing the surface roughness of the transfer material. The most successful for single-phase flow appears to be a method devised by Dipprey and Sabersky(14), and one developed by Kemeny and Cyphers(15). Dipprey and Sabersky

produced the roughness by electronically depositing copper on a rod coated with sand grains. The rod was dissolved leaving a tube of copper with close-packed sand grains on the inner surface. This method, for high Prandtl number flow, increases the h, heat transfer coefficient, by a factor of two. Figure 2.5-1. The Kemeny and Cypher method uses a helical groove in the material and a helical protrusion from the material. The grooved surface showed little improvement in the heat transfer. The helical protrusion did increase the h by up to a factor of two. Figure 2.5-2. As can be seen from Figures 2.5-1,2, the effects of the enhanced surfaces decrease with the increase in the Reynolds number. These methods also show no advantage in two-phase flow.

In two-phase flow, the heat transfer is increased by promoting nucleate boiling at as low a heat flux as possible. Young and Hummel(16) placed spots of teflon on the heated surface. This promoted nucleation as shown in Figure 2.5-3. This lowered the required level of superheat required to produce boiling. With an increase of Reynolds number, the effect of the teflon spots decreases.

The methods of increasing the heat transfer can also include the use of a smooth surface with a higher k. All of these methods of increased heat transfer would be useful in the design of the heat exchangers for the thermal cycles. What is needed is a method of comparison of these increases heat transfer surfaces to unenhanced surfaces.

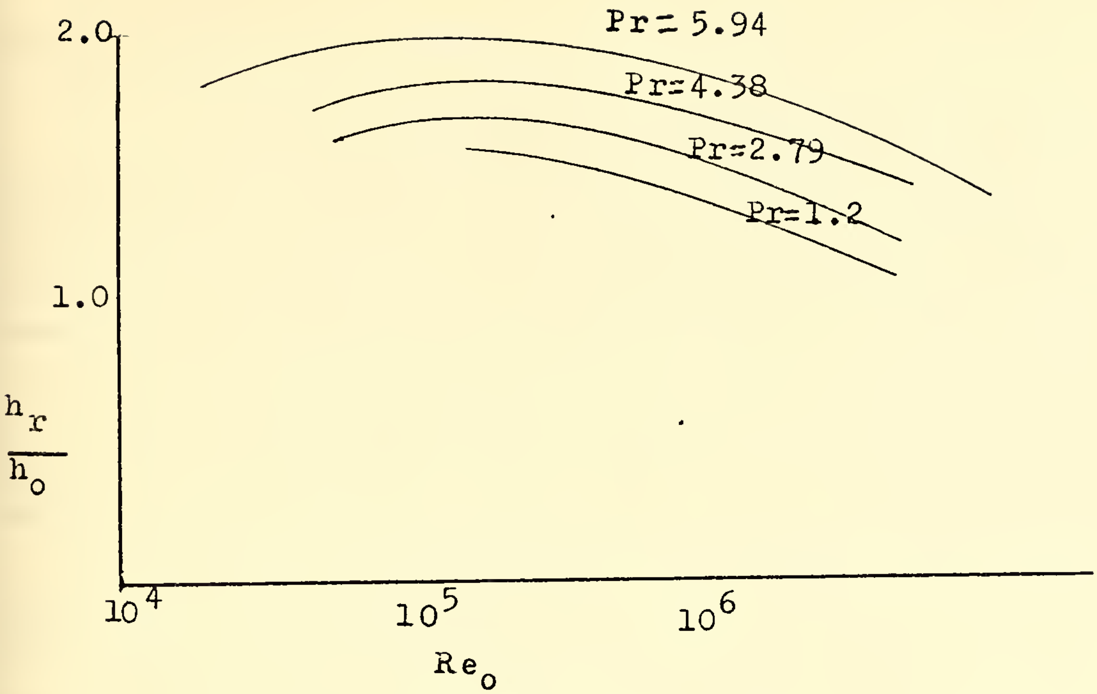


Figure 2.5-1

Dippry and Sabersky Results

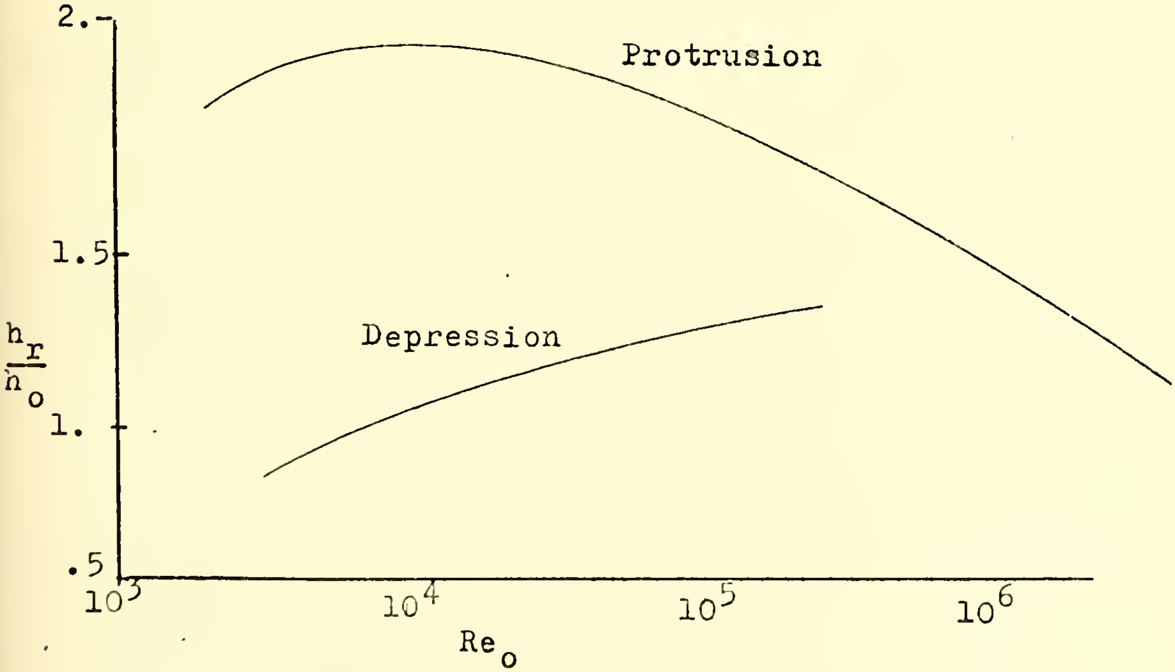


Figure 2.5-2

Kenery and Cypher Results

2.5.3 Methods of Comparison

A method of comparison has been developed by Professor W.M. Rohsenow and others working in the Heat Transfer Lab at M.I.T.(17). The data on the surfaces is presented based on the same heat transfer area and tube diameter of equivalent diameter.

$f\ Re^3$ is plotted against $j\ Re$. Figure 2.5-3.

Where: $$f\ Re^3 = P/v\ 2g\rho^2\ De^3 q/A/V\ \mu^3$$

and $$j\ Re = \frac{Ah}{V}\ \frac{Pr^{2/3}\ Deq}{A/V\ Cp\mu}$$

The following are kept constant:

g, ρ, Deq, A/V, μ, Cp, Pr

Therefore: $$f\ Re^3 \sim P/V$$

and $$j\ Re \sim \frac{Ah}{V} \text{ or } \frac{q/\Delta T_1 m}{V}$$

where $P = W\Delta P/\rho$

For the same mass flow rate and material properties:

$$A\ h/V \sim \frac{\frac{Ah}{WCp}}{V} = \frac{NTU}{V}$$

For any flow arrangement there is one NTU vers ε curve(18). Figure 2.5-4.

Where: $$\varepsilon = \frac{\Delta T \text{rise in cold fluid}}{T\text{hot in} - T\text{cold in}}$$

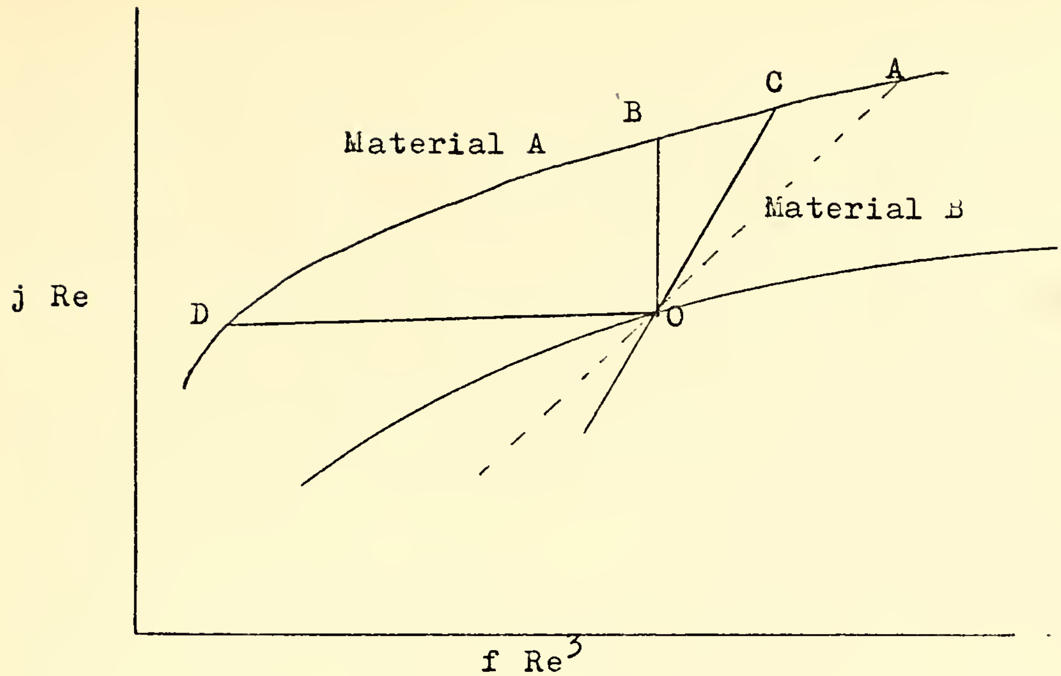


Figure 2.5-3

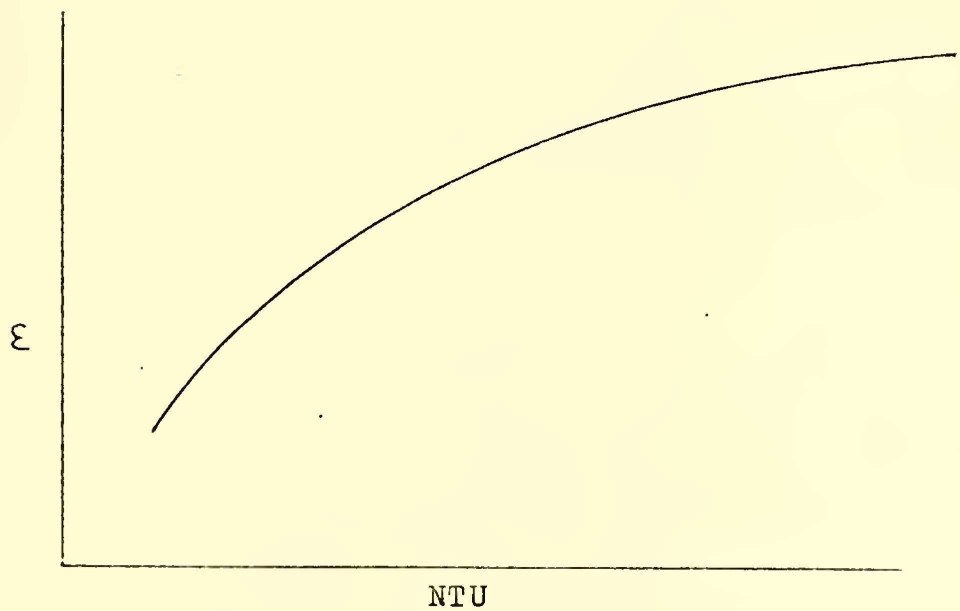


Figure 2.5-4

The two materials can now be compared at two points. See Figure 2.5-3. The mass flow rate, w, the Deq, A/V, $T_{hot\ in}$ and $T_{cold\ in}$ remain constant throughout this comparison.

At point A, tha values of w, V, and Re are the same as at point 0. This gives the same size and shape heat exchangers with different surface material. The amount of heat transfer and required pumping power will be different. These can be plotted as shown in Figure 2.5-5.

Point B has the same mass flow rate, pumping power required and volume as point 0. A heat exchanger operating at B will have the same volume as one at 0, but the shape changes and there is more heat transfer. This comparison is made in Figure 2.5-6.

If the comparison of the materials is to be made while doing the same job, point C and point 0 are examined. At these points the same heat transfer (NTU), mass flow rate and pumping power occur. The difference is that less volume is needed with material B (point C). Figure 2.5-7.

Since A/V has been held constant, the area required at point C is also less. This can be an important comparison if the heat exchanger needs to be as small as possible. A cost comparison can easily be done. As the pumping power is the same, the only variable cost is the material costs.

$$\frac{\text{Cost}_B/\text{ Square foot} \times A_B}{\text{Cost}_A/\text{ Square foot} \times A_A} = Cr$$

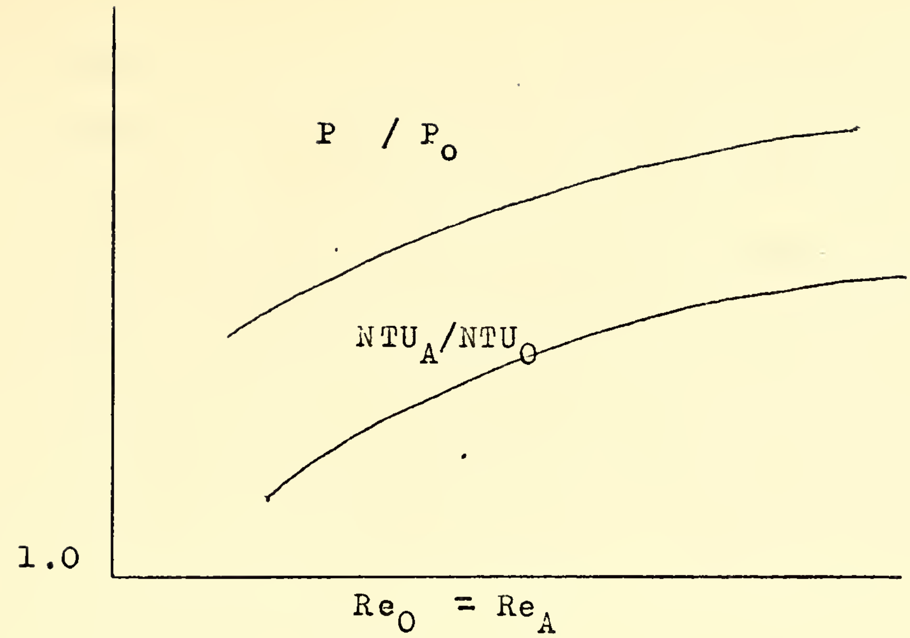


Figure 2.5-5
Comparison of Exchanger at Point O and at Point A

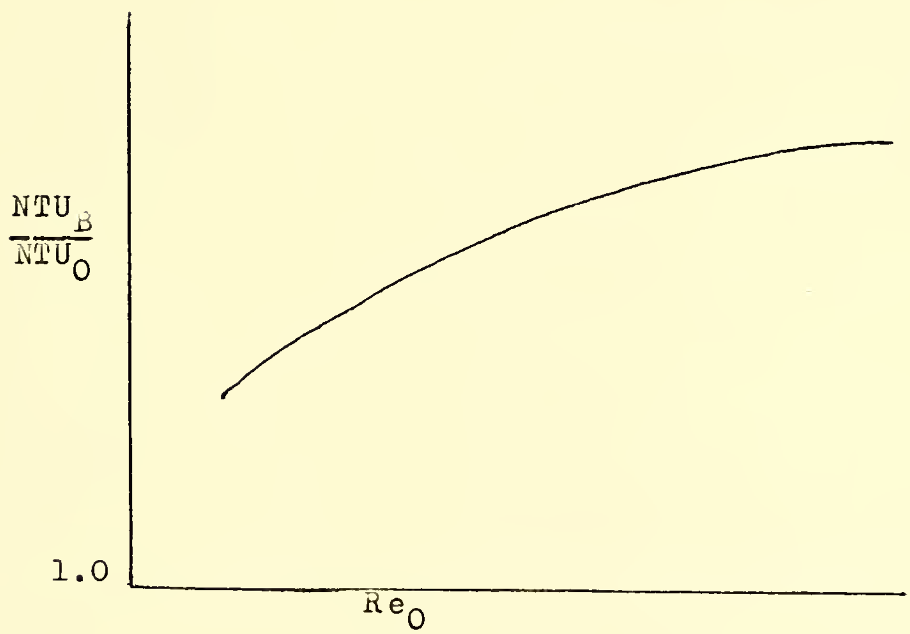


Figure 2.5-6

Comparison of Exchanger at Point O and at Point A

If Cr is greater than 1, A will be a cheaper material to use. If, however, Cr is less than 1, B is the cheaper material.

A fourth comparison can be made at point D. Here the heat transfer, mass flow rate and the volume are kept constant with point O. The shapes differ and the pumping power is lower. Figure 2.5-8. If the cost of the required pumping power was too high with material A, material B could be used to lower it.

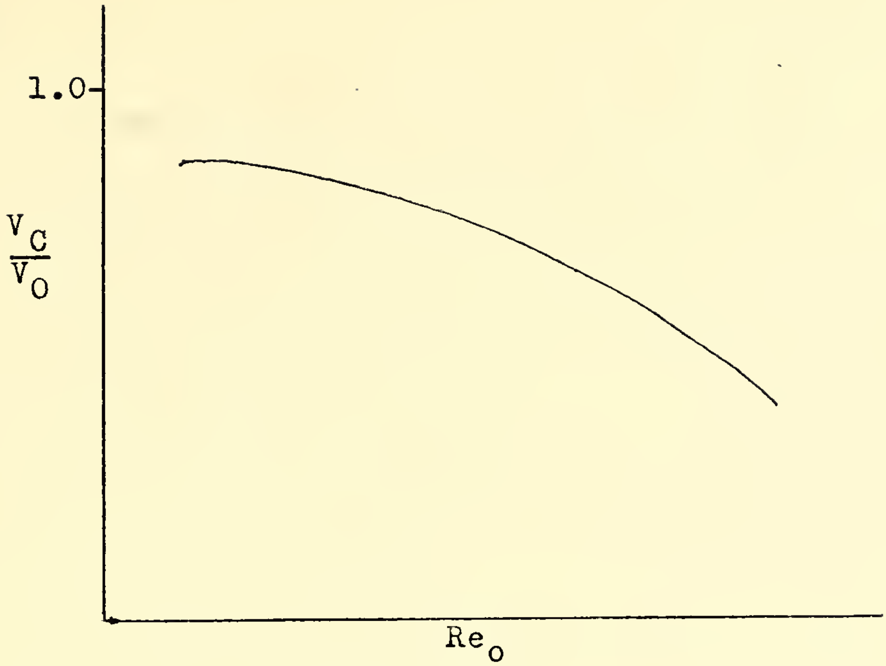


Figure 2.5-7

Comparison of Exchanger at Point O and at Point C

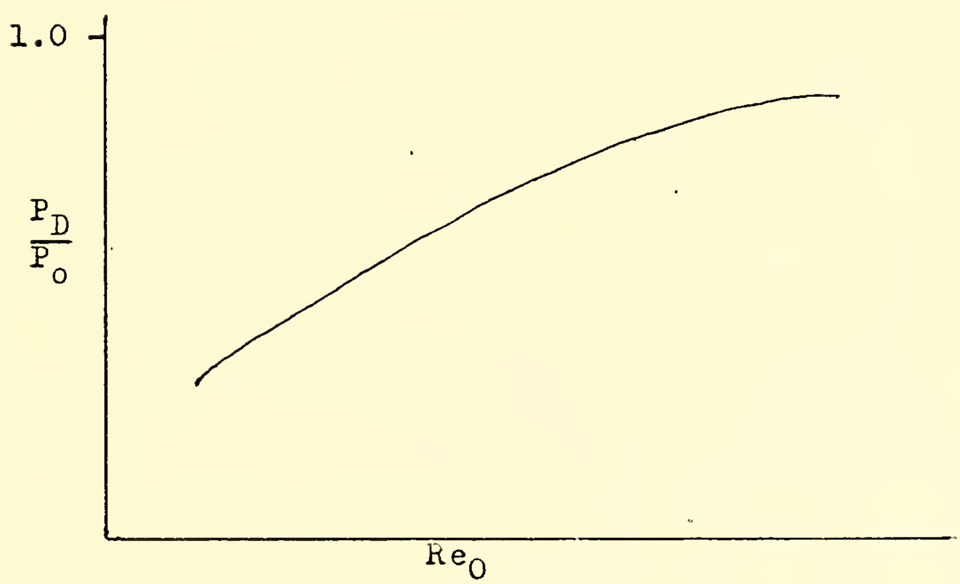


Figure 2.5-8

Comparison of Exchanger at Point O and at Point D

NOMENCLATURE

A	Heat transfer surface area
C_p	Specific heat at constant pressure
Cr	Cost ratio
Deq	Equivalent diameter
f	Friction factor
g	Gravitational acceleration
h	Local heat transfer coefficient
k	Thermal conductivity
NTU	Number of exchanger heat transfer units
P	Pumping power
ΔP	Change in pressure
Pr	Prandtl number
q	Rate of heat transfer
Re	Reynolds number
ΔT_{lm}	Logarithmic mean temperature
V	Volume of exchanger
w	Mass flow rate
ε	Exchanger heat transfer effectiveness
ρ	Mass density
μ	Absolute viscosity

Subscripts:

A	Material A
B	Material B

III. RESULTS

In this study, the objective use to develop a series of computer programs to carry out preliminary design on heat exchangers required for thermal cycles. For this development, general properties of heat exchangers were examined and mathematical models were developed for various modes of heat transfer.

The computer programs were developed and tested. All of the programs performed as desired except for the cost calculations and the author feels that they should perform successfully in studies of a broad range of inputs. The programs are explained and written in detail in the Appendix.

Two of the programs were used in detail in an examination of the proposal made by Prof. Zener. See Appendix G. The evaporation and condensation heat exchanger programs carried out the desired preliminary design and showed the variety of designs that can be carried out with these programs.

The Zener proposed cycle was examined to determine its feasibility. With the use of freon-21 as a working fluid, the evaporator must have a volume of two billion cubic feet for a 100 Mega-watt power plant. The condenser will have a minimum size of over three-hundred million cubic feet of volume. More detailed results are included in Appendix G.

IV CONCLUSIONS

The following conclusions can be drawn from the results of the study.

1. The computer can be a valuable aid in heat exchanger design work.
2. The cycle proposed by Prof. Zener is not feasible at the present time.
3. The cost analysis done in this study is very inaccurate.
4. Although the programs can give a variety of geometrical arrangements there is no way to compare these arrangements to determine the best.

V RECOMMENDATIONS

This study appears to the author to be weak in two areas. The evaluation of cost is very inaccurate. The problem of optimization was not discussed in this study. A further study of these two areas would be beneficial. As these two areas are inter-related, they could be examined together.

More examination of the cost of heat exchangers needs to be carried out. The relationships for cost caluculation derived in section 2.1 are based in 1960 data. This in in itself is a problem, but also the size of heat exchangers designed is way beyond the range of validity of these relationships. To extrapolate to 4,000 feet a curve that is valid up to 400 feet is unrealistic. More up to data values of cost should be determined. Also cost relationships that are valid for large heat exchangers should be derived. These relationships could easily be placed in the program.

The programs as written contain no optimization operations. The programs require a disigner to input various geometric arrangements and then compare the output to determine wich is best for his application. With up to data cost relationships, an optimization operation could be incorporated into the programs giving the designer the best heat exchanger within some specified limit.

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Appendix A - COUNTER-FLOW CONDENSING HEAT EXCHANGER PROGRAM

A.1 Instructions for operation and input

This program is written in FORTRAN IV and can be run as is on the Interdata computer of the Joint Civil and Mechanical Engineering Computer Facility at M.I.T. With modifications in the control, input and output cards, this program can be run on any computer that is programed to compile FORTRAN IV.

For this program, the input data is broken into two groups. The first contains four cards, three of which contain properties of the fluids and the fourth is the number, M, of input sets to follow. The second group has 2M cards. There are two cards to a set. The first card serves as an identification card and the second has the geometric arrangement of the heat exchanger; Din, Dout, S, N, KMAT, and NB. With this input arrangement, M variations of the geometry of the exchanger can be examined using the same fluid properties.

The input data is punched on the cards as follows:

(1) All values will be in units of lb_m, hr., feet, Btu, and ^{o}F.

(2) All values will be fixed point numbers with decimal points except the value of M.

(3) No commas are needed to separate the input values.

(4) The input values must be in the proper columns but need not be right or left justified.

(5) All fixed point numbers can have up to 4 (four) digits after the decimal point.

(6) The properties of the fluids, unless specified to be at a particular point, are average values over the length of the exchanger.

CARD	COLUMNS					
	1--13	14--26	27--39	40--52	53--65	66--78
1	TC1	TC2	Tw	DELHW	MKRW	ROWL
2	ROWV	MUWL	KWL	KC	PRC	MUC
	1--12	13--24	25--36	37--48	49--60	
3	ROC	HFG	CPC	PCI	PC2	
4	1--4					
5	Identification name for each geometric variation					
	1--13	14--26	27--39	40--52	53--65	66--78
6	DIN	DOUT	S	N	KMAT	NB
7	Same as Card 5					
8	Same as Card 6					

This repeats M times.

The output will appear with the identification at the top of a page and the following printed below it.

```
MASS FLOW RATE WORKING FLUID
MASS FLOW RATE COOLING FLUID
NUMBER OF TUBES
INNER DIAMETER OF TUBES
OUTER DIAMETER OF TUBES
SPACING OF TUBES
LENGTH OF TUBES
DROP IN PRESSURE COOLING FLUID
COMPARITIVE COSTS
```

For each identification there will be a separate page.

A.2 LIST OF VARIABLES

AREA	Heat transfer surface area in ft^2
COST	Cost of heat exchanger in dollars
CPC	Specific heat at constant pressure of cooling fluid in $Btu/lb_m\ ^oF$
CTB	Cost of the materials for the tubes and shell in dollars
CW	Fabrication costs in dollars
CWV	Fabrication cost per tube in dollars
DELHW	Change in enthalpy in Btu/lb_m
DIN	Inner diameter in feet
DLPC	Frictional pressure drop in cooling fluid in lb/ft^2
DOUT	Outer diameter of tubes in feet
DTLM	Log mean temperature difference in ^oF
GC	Mass flow velocity in $lb_m/hr.\ ft.^2$
HC	Heat transfer coefficient of cooling fluid in $Btu/hr.\ ft.\ ^oF$
HFG	Latent heat of condensation in Btu/lb_m
HWA	Average heat transfer coefficient of working fluid in $Btu/hr.\ ft.^2\ ^oF$
KC	Thermal conductivity of cooling fluid in $Btu/hr.\ ft.\ ^oF$
KWL	Thermal conductivity of working fluid as a liquid in $Btu/hr.\ ft.\ ^oF$
KMAT	Thermal conductivity of wall material in $Btu/hr.\ ft.\ ^oF$
LENGTH	Length of heat exchanger in feet
M	Number of groups of inputs data

MFRC	Mass flow rate of cooling fluid in lb_m/hr.
MFRW	Mass flow rate of working fluid in lb_m/hr.
MUC	Viscosity of cooling fluid in lb_m/hr. ft.
MUWL	Viscosity of working fluid as a liquid in lb_m/hr. ft.
N	Number of tubes
NB	Number of tubes in a verticle row
PCI	Inlet pressure of cooling fluid in lb/ft.^{2}
PC2	Outlet pressure of cooling fluid in lb/ft^2
PRC	Prandtl number of cooling fluid
Q	Rate of heat transfer in Btu/hr.
REC	Reynolds number of cooling fluid
ROC	Density of cooling fluid in lb_m/ft.^{3}
ROWL	Density of working fluid as a liquid in lb_m/ft.^{3}
ROWV	Density of working fluid as a vapor in lb_m/ft.^{3}
S	Spacing of tubes
TC1	Inlet temperature of cooling fluid in ^oF
TC2	Outlet temperature of cooling fluid in ^oF
TW	Temperature of working fluid in ^oF
TLF	Tube length factor in dollars/ft.
TMF	Tube material factor
U	Overall heat transfer coefficient in BTU/hr.ft^2 ^oF.

```
C     CONDENSING WITH WORKING FLUID OUTSIDE TUBES
      REAL N,MFRW,MFRC,NB,MUWL,KWL,MUC,KMAT,HWA,HFG,HC,LENGTH,KC
      READ(8,1000)TC1,TC2,TW,DELHW,MFRW,ROWL
      READ(8,1000)ROWV,MUWL,KWL,KC,PRC,MUC
      READ(8,1001)ROC,HFG,CPC,PC1,PC2
      READ(8,1002)M
      DO 50 L=1,M
      READ(3,1050)
      READ(8,1000)DIN,DOUT,S,N,KMAT,NB
1     BB=32.17*ROWL*(ROWL-ROWV)*KWL**3.*HFG
      BC=NB*DOUT*MUWL*(TW-((TC2-TC1)/2.))
      HWA=.725*(BB/BC)**.25
      HWA=9000.
      MFRC=MFRW*DELHW/(CPC*(TC2-TC1))
      GC=MFRC*4./(3.14*DIN*DIN*N)
      REC=MFRC*4./(3.14*DIN*MUC)
      IF(REC.LT.10000.) GO TO 2
      HC=.023*KC*PRC**.4/(MUC**.8*DIN**.2)*(MFRC*4./(3.14*DIN*DIN*N))
     2**.8
      GO TO 4
2     IF(REC.LT.2100.) GO TO 3
      HC=.116*KC*PRC**.33*GC**.67/DIN**.33/MUC**.67
      GO TO 4
3     HC=1.86*KC*PRC**.5*GC**.5/DIN**.5/MUC**.5
4     B=1./HC+1./HWA+(DOUT-DIN)/KMAT
      U=1./B
      Q=MFRW*DELHW
      C=(TW-TC2)/(TW-TC1)
      DTLM=((TW-TC2)-(TW-TC1))/ALOG(C)
      AREA=Q/(U*DTLM)
      LENGTH=AREA/(3.14*N*DIN)
      IF(REC.LT.2500.)GO TO 20
      DLPC=(.4*.046/REC**.2)*(MFRC*4./(3.14*DIN*DIN*N))**2.*(.5/(32.17
     1*ROC))*LENGTH
      GO TO 30
20    DLPC=128.*MUC*GC/(32.17*ROC*DIN*DIN)*LENGTH
```

```
 30   PC=PC1-PC2
      IF(DLPC.LE.PC) GO TO 40
      DD=DOUT/DIN
      DS=S/DOUT
      DIN=DIN+.01
      DOUT=DIN*DD
      S=DOUT*DS
      GO TO 1
 40   CONTINUE
C     COST CALCULATIONS
      TLF=1.3-((LENGTH-8.)**.5*0.1)
      CWV=((12.0/5.0)*(DOUT-(.5/12.0)))+3.0
C     CW IS THE FABRICATION COSTS
      CW=CWV*N
C     TMF IS THE COST FACTOR FOR THE TYPE OF MATERIAL USED
      TMF=2.5
C     CTB IS THE COST OF THE MATERIAL FOR THE TUBES AND THE SHELL
      CTB=((150.-((AREA-100.)**.5)/5.)/10.)*TLF*AREA*TMF
      COST=CW+CTB
      WRITE(5,2020)
      WRITE(5,1050)
      WRITE(5,2002) MFRW
      WRITE(5,2003) MFRC
      WRITE(5,2004) N
      WRITE(5,2005) DIN
      WRITE(5,2006) DOUT
      WRITE(5,2007) S
      WRITE(5,2008) LENGTH
      WRITE(5,2010) DLPC
      WRITE(5,2011) COST
 50   CONTINUE
 1000 FORMAT(6F13.4)
 1001 FORMAT(5F12.4)
 1002 FORMAT(I4)
 1050 FORMAT(1H1,5X,'                              ')
 2002 FORMAT(1X,'MASS FLOW RATE WORKING FLUID'5X,F12.4,1X,'LB/HR')
```

```
2003 FORMAT(1X,'MASS FLOW RATE COOLING FLUID'5X,F14.4,1X,'LB/HR')
2004 FORMAT(1X,'NUMBER OF TUBES'18X,F12.4)
2005 FORMAT(1X,'INNER DIAMETER OF TUBES'11X,F12.4,1X,'FEET')
2006 FORMAT(1X,'OUTER DIAMETER OF TUBES'11X,F12.4,1X,'FEET')
2007 FORMAT(1X,'SPACING OF TUBES'17X,F12.4,1X,'FEET')
2008 FORMAT(1X,'LENGTH OF TUBES'14X,F12.4,1X,'FEET')
2010 FORMAT(1X,'DROP IN PRESSURE COOLING FLUID'3X,F12.4,1X,'LB/SQ FT')
2011 FORMAT(1X,'COMPARITIVE COSTS'17X,F15.2,3X,'DOLLARS')
2020 FORMAT(1H1)
     END
```

Appendix B - COUNTER-FLOW EVAPORATION HEAT EXCHANGER PROGRAM

B.1 Instructions for operation and input

This program is designed to be run on the Interdata computer of the Joint Civil and Mechanical Engineering Computer Facility at M.I.T. The program is written in FORTRAN IV and can be run on any computer that is programed to compile FORTRAN IV. If run on a computer other than the Interdata, some modification may be required to the control, input and output cards.

The input data is broken into two sections. The first contains five cards, four contain fluid properties and the fifth is the number, M, of input sets to follow. The second section has 2M cards. The first card serves as an identification card and the second card has the geometric arrangement of the heat exchanger; DIN, DOUT, S, N, KMAT. With this input arrangement, M different variations of geometry can be examined using the same fluid properties.

The input data is punched on cards with the following format:

(1) All values will be in units of lb_m, hr, feet, Btu, and ^oF.

(2) All values except M will be fixed point numbers and will have decimal points. M is a floating point number.

(3) No commas are needed to separate the input values on a single card.

(4) The input values must be in the designated column but need not be right or left justified.

(5) All fixed point numbers can have up to four digits after the decimal point.

(6) The fluid properties, unless specified to be at a particular point, are average values.

CARD	COLUMN				
	1--14	15--28	29--42	43--56	57--72
1	DELHW	TH1	TH2	TSAT	ROWL
2	ROWV	MUWL	MUVW	CPWL	KWL
3	MFRW	PH1	PH2	PW1	PW2
4	HFG	CPH	MUH	ROH	KH
	1--4				
5	M				
6	Identification name is columns 5-----18				
	1--14				
7	DIN	DOUT	S	N	KMAT
8	Same as card 6				
9	Same as card 7				

This is repaeated M times

The output will appear with the identification name at the top of the page and the following printed below it.

MASS FLOW RATE WORKING FLUID
MASS FLOW RATE HEATING FLUID
NUMBER OF TUBES
INNER DIAMETER OF TUBES
OUTER DIAMETER OF TUBES
SPACING OF TUBES

(cont. overleaf)

LENGTH OF TUBES
DROP IN PRESSURE WORKING FLUID
DROP IN PRESSURE HEATING FLUID
COMPARITIVE COSTS

For each identification there will be a separate page.

B.2 LIST OF VARIABLES

AREA	Heat transfer surface area in ft^2.
COST	Cost of heat exchanger in dollars
CPH	Specific heat at constant pressure of heating fluid in $Btu/lb_m\ ^oF$
CPWL	Specific heat at constant pressure of working fluid as a liquid in $Btu/lb_m\ ^oF$
CTB	Cost of the material for the tubes and shell in dollars
CW	Cost of fabrication in dollars
CWV	Fabrication cost per tube in dollars
DELHW	Change in ethalpy of working fluid in Btu/lb_m
DELPH	Change in pressure of heating fluid in lb/ft^2.
DELPW (1)	Frictional pressure drop of working fluid per segiment of exchanger in lb/ft^2.
DELZ (1)	Length of each segiment in feet.
DELX	Change in quality in each segiment
DEQ	Equivalent diameter in feet
DIN	Inner diameter of tubes in feet
DOUT	Outer diameter of tubes in feet
FXTT	Reynolds number factor
GH	Mass flow velocity of heating fluid in $lb_m/hr.\ ft^2$.
GW	Mass flow velocity of working fluid in $lb_m/hr.\ ft^3$.
HH	Heat transfer coefficient of heating fluid in $Btu/hr.\ ft^2\ ^oF$
HFG	Latent heat of evaporation in Btu/lb_m

B.2 LIST OF VARIABLES (cont.)

HMAC	Heat transfer coefficient for macroconventive effect in Btu/hr. ft^2. ^oF
HW	Heat transfer coefficient for working fluid as a vapor is Btu/hr. ft^2. ^oF
IV	Two phase frictional multiplier
KH	Thermal conductivity of heating fluid in Btu/hr. ft. ^oF
KWL	Thermal conductivity of working fluid as a liquid in Btu/hr. ft. ^oF
M	Number of groups of input data
MFRH	Mass flow rate of heating fluid in lb_m/hr.
MFRW	Mass flow rate of working fluid in lb_m/hr.
MUH	Viscosity of heating fluid in lb_m/hr. ft.
MUWL	Viscosity of working fluid as a liquid in lb_m/hr. ft.
MUVW	Viscosity of working fluid as a vapor in lb_m/hr. ft.
N	Number of tubes in exchanger
PH1	Inlet pressure of heating fluid in lb/ft^2.
PH2	Outlet pressure of heating fluid in lb/ft^2.
PW1	Inlet pressure of working fluid in lb/ft^2.
PW2	Outlet pressure of working fluid in lb/ft^2.
PRWL	Pradtl number of working fluid as a liquid
REWL	Reynolds number of working fluid as a liquid
ROH	Density of heating fluid in lb_m/ft^3.
ROWL	Density of working fluid as a liquid in lb_m/ft^3.

B.2 LIST OF VARIABLES (cont).

ROWV	Density of working fluid as a vapor in lb_m/ft^3.
REH	Reynolds number of heating fluid
S	Spacing between tubes
TH1	Inlet temperature of heating fluid in ^oF
TH2	Outlet temperature of heating fluid in ^{o} F
TLF	Tube length factor
TMF	Tube material factor
TSAT	Saturation temperature in ^oF
X	Quality
XTT	Martinelli parameter
U	Overall heat transfer coefficient in Btu/hr. ft^2 ^oF
Z	Length of heat exchanger in ft.

```
C     EVAPORATION  WORKING FLUID INSIDE TUBES
      DIMENSION DELZ(6),DELPW(6)
      REAL MUWL,MUWV,KWL,MFRW,MFRH,N,IV,MUH,KH
      READ(8,1000)DELHW,TH1,TH2,TSAT,ROWL
      READ(8,1000)ROWV,MUWL,MUWV,CPWL,KWL
      READ(8,1000)MFRW,PH1,PH2,PW1,PW2
      READ(8,1000)HFG,CPH,MUH,ROH,KH
      READ(8,1001)M
      DO 40 L=1,M
      READ(8,1050)
      READ(8,1000)DIN,DOUT,S,N,KMAT
 1    DEQ=(1.271*(S/DOUT)**2.-1.)*DOUT
      MFRH=MFRW*DELHW/(CPH*(TH1-TH2))
      GW=MFRW/((3.14/4.)*DIN*DIN*N)
      PRWL=CPWL*MUWL/KWL
      A=(MUWL/MUWV)**.1
      RC=(ROWV/ROWL)**.5
      DELX=.2
      X=.1
      DO 10 I=1,5
      XTT=A*RC*((1.-X)**.9)
      REWL=GW*DIN*(1.-X)/MUWL
      FXTT=0.15*((1./XTT)+(2.85/(XTT**.476)))
      IV=1.+2.85*(XTT**0.522)
 3    HMAC=KWL*FXTT*0.023*PRWL**.4*REWL**.8/DIN
      GH=MFRH/(.785*DEQ*DEQ*N)
      HH=0.023*KH*((CPH*MUH/KH)**.4)*GH**.8/(MUH**.8*DEQ**.2)
      DELT=TH1-TH2
      T=DELT/5.
      D=I
      TW=TH2+(D*T)
      BBB=1./HH+1./HMAC
      U=1./BBB
      AREA=MFRW*DELHW*.2/(U*(TW-TSAT))
      DELZ(I)=AREA/(3.14*DIN*N)
      IF(REWL.LT.2500.)GO TO 4
```

```
      CPDZ=IV*IV*(0.09*(MUWV**.2))*(GW**1.8)*(X**1.8)/(32.17*RCWV*DIN
     2**1.2)
      GO TO 5
4     CPDZ=IV*IV*128.*MUWV*GW*X**1.8/(DIN*DIN*32.17*ROWV)
5     DELPW(I)=CPDZ*DELZ(I)
      X=X+.2
10    CONTINUE
      REH=GH*DEQ/MUH
      IF(REH.LT.2500.)GO TO 11
      DELPH=4.*.046/REH**.2*(MFRH*4./(3.14*DEQ*DEQ*N))**2.*.5/(32.17*
     1RCH)
      GO TO 12
11    DELPH=128.*MUH*GH/(32.17*ROH*DEQ*DEQ)
12    DEPW=0.
      Z=0.
      DO 13 I=1,5
      DEPW=DEPW+DELPW(I)
      Z=Z+DELZ(I)
13    CONTINUE
      DEPH=DELPH*Z
      PH=PH1-PH2
      PW=PW1-PW2
      IF(DEPW.LE.PW) GO TO 20
14    DD=DIN/DOUT
      DS=S/DOUT
      DIN=DIN+.1
      DOUT=DIN/DD
      S=DS*DOUT
      GO TO 1
20    IF(DEPH.LE.PH) GO TO 30
      GO TO 14
30    CONTINUE
      AREA=3.14*DOUT*Z*N
C     COST CALCULATIONS
      IF(DOUT.LE.0.042)CWV=3.0
      TLF=1.3-((Z-8.)**.5*0.1)
```

```
      CWV=((12.0/5.0)*(DOUT-(.5/12.0)))+3.0
C     CW IS THE FABRICATION COSTS
      CW=CWV*N
C     TMF IS THE COST FACTOR FOR THE TYPE OF MATERIAL USED
      TMF=2.5
C     CTB IS THE COST OF THE MATERIAL FOR THE TUBES AND THE SHELL
      CTB=((150.-((AREA-100.)**.5)/5.)/10.)*TLF*AREA*TMF
      COST=CW+CTB
      WRITE(5,2020)
      WRITE(5,1050)
      WRITE(5,2002) MFRW
      WRITE(5,2003) MFRH
      WRITE(5,2004) N
      WRITE(5,2005) DIN
      WRITE(5,2006) DOUT
      WRITE(5,2007) S
      WRITE(5,2008) Z
      WRITE(5,2009) DEPW
      WRITE(5,2010) DEPH
      WRITE(5,2011) COST
40    CONTINUE
1000  FORMAT(5F14.4)
1001  FORMAT(I4)
1050  FORMAT(5X,'                              ')
2002  FORMAT(1X,'MASS FLOW RATE WORKING FLUID'5X,F12.4,1X,'LB/HR')
2003  FORMAT(1X,'MASS FLOW RATE HEATING FLUID'3X,F14.4,1X,'LB/HR')
2004  FORMAT(1X,'NUMBER OF TUBES'18X,F12.4)
2005  FORMAT(1X,'INNER DIAMETER OF TUBES'11X,F12.4,1X,'FEET')
2006  FORMAT(1X,'OUTER DIAMETER OF TUBES'11X,F12.4,1X,'FEET')
2007  FORMAT(1X,'SPACING OF TUBES'17X,F12.4,1X,'FEET')
2008  FORMAT(1X,'LENGTH OF TUBES'14X,F12.4,1X,'FEET')
2009  FORMAT(1X,'DROP IN PRESSURE WORKING FLUID'3X,F12.4,1X,'LB/SQ FT')
2010  FORMAT(1X,'DROP IN PRESSURE HEATING FLUID'3X,F12.4,1X,'LB/SQ FT')
2011  FORMAT(1X,'COMPARITIVE COSTS'15X,F14.2,3X,'DOLLARS')
2020  FORMAT(1H1)
      END
```

Appendix C. COUNTER FLOW HEATER HEAT EXCHANGER PROGRAM

C.1 Instructions for Operation and Input

This program is written in FORTRAN IV and can be run as is on the Interdata computer of the Joint Civil and Mechanical Engineering Computer Facility at M.I.T. The program can be run on any computer facility that is designed with the ability to compile FORTRAN IV. If run on a computer other than the Interdata, some modifications may be required to the control, input and output cards.

The input data is broken into two sections. The first section contains eight input data cards. The first seven of these cards contain the fluid properties and the eight is the number, M, of input sets in the second section. The second section has 2M cards. The first card in each of these M sets serves as an identification card. The second card has the data on the geometric arrangement of the heat exchanger; DIN, DOUT, and S. With this input arrangement, M different geometrical arrangements can be examined using the same fluid properties.

The input data is punched on cards as follows:

(1) All values will be in units of lb_m, hr., feet, Btu and ^oF.

(2) All values except M will be fixed point numbers and will have a decimal point. M is a floating point number.

(3) No commas are needed to separate the input data on a single card.

(4) The input values must be in the designated column but need not be right or left justified.

(5) All fixed point numbers can have up to 4 (four) digits after the decimal point.

(6) The fluid properties, unless specified to be at a particular point, are average values. Any property that the last figure of its variable name is a number is specified to be at that temperature.

CARD	COLUMN							
	1--12	13--24	25--36	37--48	49--60			
1	MFRW	TH1	TH2	TW1	TW2			
2	PH1	PH2	PW1	PW2	MUH			
3	MUW	CPH	CPW	ROH	ROW			
4	KH	KW	DCPW1	DCPW2	DCPW3			
	1--10	11--20	21--30	31--40	41--50	51--60	61--70	71--80
5	DCPH1	DCPH2	DCPH3	TW3	TH3	DMUW1	DMUW2	DMUW3
6	DMUH1	DMUH2	DMUH3	DKW1	DKW2	DKW3	DKH1	DKH2
7	DKH3	DROW1	DROW2	DROW3	DROH1	DROH2	DROH3	
	1--4							
8	M							
	5--------18							
9	Identification name							
	1--12	13--24	25--36					
10	DIN	DOUT	S					
11	Same as 9							
12	Same as 10							

This repeats M times

(cont. overleaf)

The output will appear with the identification name at the top of a page and the following printed below it.

```
MASS FLOW RATE WORKING FLUID
MASS FLOW RATE HEATING FLUID
NUMBER OF TUBES
INNER DIAMETER OF TUBES
OUTER DIAMETER OF TUBES
SPACING OF TUBES
LENGTH OF TUBES
DROP IN PRESSURE WORKING FLUID
DROP IN PRESSURE HEATING FLUID
COMPARITIVE COSTS
```

For each identification there will be a separate page.

LIST OF VARIABLES

AREA	Heat transfer area in ft^2.
COST	Cost of heat exchanger in dollars
CPH	Specific heat at constant pressure of heating fluid in $Btu/lb_m\ ^oF$
CPW	Specific heat at constant pressure of working fluid in $Btu/lb_m\ ^oF$
CTB	Cost of materials for shell and tubes in dollars
CW	Fabrication costs in dollars
CWV	Fabrication costs per tube in dollars
DIN	Inner diameter of tubes in ft.
DOUT	Outer diameter of tubes in ft.
DEQ	Equivalent diameter in ft.
DPH	Frictional pressure drop in heating fluid in lb/ft^2.
DPW	Frictional pressure drop of working fluid in lb/ft^2.
DTLM	Log mean temperature difference in ^oF
D___1	Property ____ at temperature 1
D___2	Property ____ at temperature 2
D___3	Property ____ at temperature 3
GH	Mass flow velocity of heating fluid in $lb_m/hr.\ ft^2$.
GW	Mass flow velocity of working fluid in $lb_m/hr.\ ft^2$.
KH	Thermal conductivity of heating fluid in $Btu/hr.\ ft.\ ^oF$
KW	Thermal conductivity of working fluid in $Btu/hr.\ ft.\ ^oF$
LENGTH	Length of heat exchanger in ft.

LIST OF VARIABLES (cont).

MFRH	Mass flow rate of heating fluid in lb_m/hr
MFRW	Mass flow rate of working fluid in lb_m/hr.
MUH	Viscosity of heating fluid in $lb_m/hr.\ ft$.
MUW	Viscosity of working fluid in $lb_m/hr.\ ft$.
M	Number of input data groups
N	Number of tubes
PH1	Inlet pressure of heating fluid in lb/ft^2.
PH2	Outlet pressure of heating fluid in lb/ft^2.
PW1	Inlet pressure of working fluid in lb/ft^2.
PW2	Outlet pressure of working fluid in lb/ft^2.
PART	Temperature ratio
PRH	Prandtl number of heating fluid
PRW	Prandtl number of working fluid
REH	Reynolds number of heating fluid
REW	Reynolds number of working fluid
ROH	Density of heating fluid in lb_m/ft^3.
ROW	Density of working fluid in lb_m/ft^3.
S	Tube spacing
TH1	Inlet temperature of heating fluid in ^oF
TH2	Outlet temperature of heating fluid in ^oF
TH3	Intermediate temperature of heating fluid in ^oF
TW1	Inlet temperature of working fluid in ^oF
TW2	Outlet temperature of working fluid in ^oF
TW3	Intermediate temperature of working fluid in ^oF

LIST OF VARIABLES (cont).

TW1	Inlet temperature of working fluid in F
TW2	Outlet temperature of working fluid in F
TW3	Intermediate temperature of working fluid in F
TLF	Tube length factor
TMF	Tube material factor
X1------X9	are grouped constants
XXL----XX12	are grouped constants
X------(J)	Above properties for each segment of exchanger

```
C     .COUNTER FLOW HEATER WITH WORKING FLUID OUTSIDE TUBES
       DIMENSION XLN(5),XAREA(5),XDPH(5),XDPW(5)
       DIMENSION XTH1(5),XTH2(5),XTW1(5),XTW2(5),XPH1(5),XPH2(5),XPW1(5)
       DIMENSION XPW2(5),XKH(5),XKW(5),XMUH(5),XMUW(5),XCPH(5),XCPW(5)
       DIMENSION XROW(5),XROH(5),XPRW(5),XPRH(5)
       REAL MFRW,MFRH,MUW,MUH,KH,KW,N,LENGTH
       READ(8,1000)MFRW,TH1,TH2,TW1,TW2
       READ(8,1000)PH1,PH2,PW1,PW2,MUH
       READ(8,1000)MUW,CPH,CPW,ROH,ROW
       READ(8,1000)KH,KW,DCPW1,DCPW2,DCPW3
       READ(8,1001)DCPH1,DCPH2,DCPH3,TW3,TH3,DMUW1,DMUW2,DMUW3
       READ(8,1001)DMUH1,DMUH2,DMUH3,DKW1,DKW2,DKW3,DKH1,DKH2
       READ(8,1002)DKH3,DROW1,DROW2,DROW3,DROH1,DROH2,DROH3
       READ(8,1003)M
       DO 60 L=1,M
       READ(8,1050)
       READ(8,1005)DIN,DOUT,S
 1     DEQ=(1.271*(S/DOUT)**2.-1.)*DOUT
       MFRH=MFRW*CPW*(TW2-TW1)/(CPH*(TH1-TH2))
       REW=DEQ*MFRW*4./(3.14*DEQ*DEQ)/MUW
       REH=DIN*MFRH*4./(3.14*DIN*DIN)/MUH
       PRW=CPW*MUW/KW
       PRH=CPH*MUH/KH
       IF(REH.LT.2500.)GO TO 4
       X1=.092*MUH**.2/(32.17*ROH)
       GO TO 5
 4     X1=32.*MUH/(32.17*ROH**3.*REH)
 5     IF(REW.LT.2500.)GO TO 6
       X2=.092*MUW**.2/(32.17*ROW)
       GO TO 12
 6     X2=32.*MUW/(32.17*ROW**3.*REW)
 12    IF(REH.LT.10000.) GO TO 32
       X3=.023*KH*(CPH*MUH/KH)**.4/MUH**.8
       GO TO 24
 32    IF(REH.LT.2100.)GO TO 23
       X3=.166*KH*PRH**.33/(REH**.33*MUH**.8)
```

```
      GO TO 24
23    X3=1.36*KH*PRH**.5/(REH**.13*MUH**.8)
24    IF(REW.LT.10000.) GO TO 25
      X4=.023*KW*(CPW*MUW/KW)**.4/MUW**.8
      GO TO 7
25    IF(REW.LT.2100.)GO TO 26
      X4=.106*KW*PRW**.33/(REW**.13*MUW**.8)
      GO TO 7
26    X4=1.86*KW*PRW**.5/(REW**.13*MUW**.8)
7     X5=(HH1-HH2)/(PW1-PW2)
      X6=(MFRH*DOUT/MFRW/DIN)**.4*(X2*X5/X1)**.33
      X7=X3/X4*(X2*X5/X1)**.33*(DOUT/DIN)**.2*(MFRH/MFRW)**.2
      X8=1./X3*(1.+DIN*X7/DOUT)
      B=(TH1-TW2)/(TH2-TW1)
      DTLN=(TH1-TW2)-(TH2-TW1)/ALOG(B)
      X9=CPH*(TH1-TH2)/(4.*DTLM)
      GH=((PH1-PH2)/(X1*X8*X9))**.5
      GW=GH/X6
      D=MFRH*4./(GH*3.14*DIN*DIN)
      K=D
      A=FLOAT K
      DT=TW2-TW1
      DDT=DT/5.
      DTW1=TW1
      DO 100 J=1,5
      XTW1(J)=DTW1
      XTW2(J)=XTW1(J)+DDT
      DTW1=XTW2(J)
100   CONTINUE
      DO 110 J=1,5
      TM=(XTW2(J)-XTW1(J))/2.+XTW1(J)
      IF(TM.GT.TC3)GO TO 101
      PART=(TW3-TM)/(TW3-TW1)
      XCPW(J)=PART*(DCPW3-DCPW1)+DCPW1
      XKW(J)=PART*(XKW3-XKW1)+XKW1
      XMUW(J)=PART*(DMUW3-DMUW1)+DMUW1
```

```
      XROW(J)=PART*(DROW3-DROW1)+DROW1
      GO TO 102
101   PART=(TW2-TM)/(TW2-TW3)
      XCPW(J)=PART*(DCPW2-DCPW3)+DCPW3
      XKW(J)=PART*(DKW2-DKW3)+DKW3
      XMUW(J)=PART*(DMUW2-DMUW3)+DMUW3
      XROW(J)=PART*(DROW2-DROW3)+DROW3
102   F=1.
110   CONTINUE
      DTH1=TH1
      DO 120 J=1,4
      XTH1(J)=DTH1
      DTH=MFRW*XCPW(J)*(XTW2(J)-XTW1(J))/(MFRH*CPH)
      XTH2(J)=XTH1(J)-DTH
      DTH1=XTH2(J)
120   CONTINUE
      J=5
      XTH2(J)=TH2
      XTH1(J)=DTH1
      DO 130 J=1,5
      TM=(XTH1(J)-XTH2(J))/2.+XTH2(J)
      IF(TM.LT.TW3)GO TO 121
      PART=(TM-TH3)/(TH1-TH3)
      XCPH(J)=PART*(DCPH1-DCPH3)+DCPH3
      XKH(J)=PART*(DKH1-DKH3)+DKH3
      XMUH(J)=PART*(DMUH1-DMUH3)+DMUH3
      XROH(J)=PART*(DROH1-DROH3)+DROH3
      GO TO 122
121   PART=(TM-TH2)/(TH3-TH2)
      XCPH(J)=PART*(DCPH3-DCPH2)+DCPH2
      XKH(J)=PART*(DKH3-DKH2)+DKH2
      XMUH(J)=PART*(DMUH3-DMUH2)+DMUH2
      XROH(J)=PART*(DROH3-DROH2)+DROH2
122   F=1.
130   CONTINUE
      DO 140 J=1,5
```

```
      XPW1(J)=PW1+((XTW1(J)-TW1)/(TW2-TW1))*(PW1-PW2)
      XPW2(J)=PW1+((XTW2(J)-TW1)/(TW2-TW1))*(PW1-PW2)
      XPH1(J)=PH1+((TH1-XTH1(J))/(TH1-TH2))*(PH1-PH2)
      XPH2(J)=PH1+((TH1-XTH2(J))/(TH1-TH2))*(PH1-PH2)
140   CONTINUE
      DO 20 I=1,5
      XPRH(I)=XCPH(I)*XMUH(I)/XKH(I)
      XPRW(I)=XCPW(I)*XMUW(I)/XKW(I)
11    IF(REH.LT.2500.) GO TO 14
      XX1=.092*XMUH(I)**.2/(32.17*XRCH(I))
      GO TO 15
14    XX1=32.*XMUH(I)/(32.17*XRCH(I)**3.*REH)
15    IF(REW.LT.2500.) GO TO 16
      XX2=.092*XMUW(I)**.2/(32.17*XRCW(I))
      GO TO 41
16    XX2=32.*XMUW(I)/(32.17*XRCW(I)**3.*REW)
41    IF(REH.LT.10000.) GO TO 42
      XX3=.023*XKH(I)*(XCPH(I)*XMUH(I)/XKH(I))**.4/XMUH(I)**.8
      GO TO 44
42    IF(REH.LT.2100.) GO TO 43
      XX3=.116*XKH(I)*XPRH(I)**.33/(REH**.33*XMUH(I)**.8)
      GO TO 44
43    XX3=1.86*XKH(I)*XPRH(I)**.5/(REH**.13*XMUH(I)**.8)
44    IF(REW.LT.10000.)GO TO 45
      XX4=.023*XKW(I)*(XCPW(I)*XMUW(I)/XKW(I))**.4/XMUW(I)**.8
      GO TO 17
45    IF(REW.LT.2100.) GO TO 46
      XX4=.116*XKW(I)*XPRW(I)**.33/(REW**.33*XMUW(I)**.8)
      GO TO 17
46    XX4=1.86*XKW(I)*XPRW(I)**.5/(REW**.13*XMUW(I)**.8)
17    XX5=(XPH1(I)-XPH2(I))/(XPW1(I)-XPW2(I))
      XX6=(MFRH*DOUT/MFRW/DIN)**.4*((XX2*XX5/XX1)**.33)
      XX7=XX3/XX4*(XX2*XX5/XX1)**.33*(DOUT/DIN)**.2*(MFRH/MFRW)**.2
      XX8=1./XX3*(1.+DIN*XX7/LCUT)
      BB=(XTH1(I)-XTW2(I))/(XTH2(I)-XTW1(I))
      DTLM=(XTH1(I)-XTW2(I))-(XTH2(I)-XTW1(I))/ALOG(BB)
```

```
      XX9=XCPH(I)*(XTH1(I)-XTH2(I))/(4.*DTLM)
      XX10=GH
      XX11=N*DIN*DIN
      XX12=3.14*XX8*XX9*XX10**.2*XX11
      XLN(I)=XX9*XX8*GH**.2*DIN**1.2
      XAREA(I)=XX12*DIN**.2
      XDPH(I)=XX1*GH**1.8*XLN(I)/(DIN**1.2)
      XDPW(I)=XX2*GW**1.8*XLN(I)/(DEQ**1.2)
   20 CONTINUE
      LENGTH=0.
      AREA=0.
      DPH=0.
      DPW=0.
      DO 21 I=1,5
      LENGTH=LENGTH+XLN(I)
      AREA=AREA+XAREA(I)
      DPH=DPH+XDPH(I)
      DPW=DPW+XDPW(I)
   21 CONTINUE
C     COST CALCULATIONS
      IF(DOUT.LE.0.042)CWV=3.0
      TLF=1.3-((LENGTH-8.)**.5*0.1)
      CWV=((12.0/5.0)*(DOUT-(.5/12.0)))+3.0
C     CW IS THE FABRICATION COSTS
      CW=CWV*N
C     TMF IS THE COST FACTOR FOR THE TYPE OF MATERIAL USED
C     CTB IS THE COST OF THE MATERIAL FOR THE TUBES AND THE SHELL
      TMF=2.5
      CTB=((150.-((AREA-100.)**.5)/5.)/10.)*TLF*AREA*TMF
      COST=CW+CTB
      WRITE(5,2020)
      WRITE(5,1050)
      WRITE(5,2002) NFRW
      WRITE(5,2003) NFRH
      WRITE(5,2004) N
      WRITE(5,2005) DIN
```

```
      WRITE(5,2006) DELT
      WRITE(5,2007) S
      WRITE(5,2008) LENGTH
      WRITE(5,2009) DEPW
      WRITE(5,2010) DPH
      WRITE(5,2011) COST
60    CONTINUE
1000  FORMAT(5F12.4)
1001  FORMAT(9F10.4)
1002  FORMAT(7F10.4)
1003  FORMAT(I4)
1005  FORMAT(3F12.4)
1050  FORMAT(5X,'                             ')
2002  FORMAT(1X,'MASS FLOW RATE WORKING FLUID'5X,F12.4,1X,'LB/HR')
2003  FORMAT(1X,'MASS FLOW RATE HEATING FLUID'5X,F12.4,1X,'LB/HR')
2004  FORMAT(1X,'NUMBER OF TUBES'18X,F12.4)
2005  FORMAT(1X,'INNER DIAMETER OF TUBES'11X,F12.4,1X,'FEET')
2006  FORMAT(1X,'OUTER DIAMETER OF TUBES'11X,F12.4,1X,'FEET')
2007  FORMAT(1X,'SPACING OF TUBES'17X,F12.4,1X,'FEET')
2008  FORMAT(1X,'LENGTH OF TUBES'14X,F12.4,1X,'FEET')
2009  FORMAT(1X,'DROP IN PRESSURE WORKING FLUID'3X,F12.4,1X,'LB/SQ FT')
2010  FORMAT(1X,'DROP IN PRESSURE HEATING FLUID'3X,F12.4,1X,'LB/SQ FT')
2011  FORMAT(1X,'COMPARITIVE COSTS'17X,F12.2,3X,'DOLLARS')
2020  FORMAT(1H1)
      END
```

Appendix D. COUNTER-FLOW COOLER HEAT EXCHANGER PROGRAM

D.1 Instructions for Operation and Inputs

This program is written in FORTRAN IV and can be run as is on the Interdata computer of the Joint Civil and Mechanical Engineering Computer Facility at M.I.T. The program can be run on any computer facility that is designed to compile FORTRAN IV. If run on a computer other than the Interdata, some modifications may be required to the control, imput and output cards.

The input deck is broken into two sections. The first contains eight input cards. The first seven cards contain the fluid properties and the eighth is the number, M, of input sets in the second section. The second section has 2M cards. The first card of each set serves as an identification card. The second card contains the data on the geometric arrangement of the heat exchanger; DIN, DOUT, and S. With this arrangement of the input data M different geometrical arrangement can be examined using the same fluid properties.

The input data is punched on cards as follows:

(1) All values will be in units of lb_m, hr., feet, Btu and ^oF

(2) All values except M will be fixed point numbers and will have a decimal point. M is a floating point number

(3) No commas are needed to separate the input data on a single card

(4) The input values must be in the designated column but do not need to be right or left justified

(5) All fixed point numbers can have up to 4 (four) digits after the decimal point.

(6) The fluid properties, unless specified to be that at a particular point, are average values. Any property that the last figure of its variable name is a number is specified to be at that temperature.

CARD	COLUMN							
	1--12	13--24	25--36	37--48	49--60			
1	TC1	TC2	TW1	TW2	PW1			
2	PW2	PC1	PC2	MFRW	MUW			
3	CPW	MUC	CPC	KW	KC			
4	ROW	ROC	DCPW1	DCPW2	DCPW3			
	1--10	11--20	21--30	31--40	41--50	51--60	61---70	71--80
5	DCPC1	DCPC2	DCPD3	TW3	TC3	DMUW1	DMUW2	DMUW3
6	DMUC1	DMUC2	DMUC3	DKW1	DKW2	DKW3	DKC1	DKC2
7	DKC3	DROW1	DROW2	DROW3	DROC1	DROW2	DROW3	
	1---4							
8	M							
	5--------18							
9	Identification name							
	1--12	13--24	25--36					
10	DIN	DOUT	S					
11	Same as 9							
12	Same as 10							

These last two cards are repeated M times

(cont. overleaf)

The output will appear with the identification name at the top of a page and the following printed below it.

```
MASS FLOW RATE WORKING FLUID
MASS FLOW RATE COOLING FLUID
NUMBER OF TUBES
INNER DIAMETER OF TUBES
OUTER DIAMETER OF TUBES
SPACING OF TUBES
LENGTH OF TUBES
DROP IN PRESSURE WORKING FLUID
DROP IN PRESSURE COOLING FLUID
COMPARITIVE COST
```

For each identification name there will be a separate page.

LIST OF VARIABLES

AREA	Heat transfer area in ft^2.
CPC	Specific heat at constant pressure for cooling fluid in $Btu/lb_m\ ^oF$
CPW	Specific heat at constant pressure of working fluid in $Btu/lb_m\ ^oF$
COST	Cost of heat exchanger in dollars
CTB	Cost of materials for shell and tubes in dollars
CW	Fabrication costs in dollars
CWV	Fabrication costs per tube in dollars
DIN	Inner diameter of tubes in ft.
DOUT	Outer diameter of tubes in ft.
DEQ	Equivalent diameter in ft.
DPC	Frictional pressure drop of cooling fluid in lb/ft^2.
DPW	Frictional pressure drop of working fluid in lb/ft^2.
DTLM	Log mean temperature in ^oF
D___1	Property______ at temperature 1
D___2	Property______ at temperature 2
D___3	Property______ at temperature 3
GC	Mass flow velocity of cooling fluid in $lb_m/hr.\ ft^2$.
GW	Mass flow velocity of working fluid in $lb_m/hr.\ ft^2$.
KC	Thermal conductivity of cooling fluid in Btu/hr. ft. ^oF
KW	Thermal conductivity of working fluid in Btu/hr. ft. ^oF
LENGTH	Length of heat exchanger in ft.

LIST OF VARIABLES (cont).

M	Number of inputs data groups
MFRC	Mass flow rate of cooling fluid in lb_m/hr.
MFRW	Mass flow rate of working fluid in lb_m/hr.
MUC	Viscosity of cooling fluid in lb_m/hr. ft.
MUW	Viscosity of working fluid in lb_m/hr. ft.
N	Number of tubes in exchanger
PC1	Inlet pressure of cooling fluid in lb/ft^2.
PC2	Outlet pressure of cooling fluid in lb/ft^2.
PW1	Inlet pressure of working fluid in lb/ft^2.
PW2	Outlet pressure of working fluid in lb/ft^2.
PRC	Prandtl number of cooling fluid
PRW	Prandtl number of working fluid
PART	Temperature ratio
REC	Reynolds number of cooling fluid
REW	Reynolds number of working fluid
ROC	Density of cooling fluid in lb_m/ft^3.
ROW	Density of working fluid in lb_m/ft^3.
S	Spacing of tubes
TC1	Inlet temperature of cooling fluid in ^oF
TC2	Outlet temperature of cooling fluid in ^oF
TC3	Intermediate temperature of cooling fluid in ^oF
TLF	Tube length factor
TMF	Tube material factor

LIST OF VARIABLES (cont).

TW1	Inlet temperature of working fluid in ^{o}F
TW2	Outlet temperature of working fluid in ^{o}F
TW3	Intermediate temperature of working fluid in ^{o}F
X1-----X9	are grouped constants
XX1----XX12	are grouped constants
X------(J)	Above properties for each segment of exchanger

```
C     COUNTER FLOW COOLER WITH WORKING FLUID OUTSIDE TUBES
      DIMENSION XLN(5),XAREA(5),XDPC(5),XDPW(5)
      DIMENSION XTC1(5),XTC2(5),XTW1(5),XTW2(5),XPC1(5),XPC2(5),XPW1(5)
      DIMENSION XPW2(5),XKC(5),XKW(5),XMUC(5),XMUW(5),XCPC(5),XCPW(5)
      DIMENSION XFCW(5),XRCC(5),XPRW(5),XPRC(5)
      REAL MFRW,MFRC,KMAT,MUW,MUC,KW,KC,N,LENGTH
      READ(8,1000)TC1,TC2,TW1,TW2,PW1
      READ(8,1000)PW2,PC1,PC2,MFRW,MUW
      READ(8,1000)CPW,MUC,CPC,KW,KC
      READ(8,1000)ROW,ROC,DCPW1,DCPW2,DCPW3
      READ(8,1001)DCPC1,DCPC2,DCPC3,TW3,TC3,DMUW1,DMUW2,DMUW3
      READ(8,1001)DMUC1,DMUC2,DMUC3,DKW1,DKW2,DKW3,DKC1,DKC2
      READ(8,1002)DKC3,DROW1,DROW2,DROW3,DROC1,DROC2,DROC3
      READ(8,1003)M
      DO 60 L=1,M
      READ(8,1050)
      READ(8,1005)DIN,DOUT,S
1     DEQ=(1.271*(S/DOUT)**2.-1.)*DOUT
      MFRC=MFRW*CPW*(TW1-TW2)/(CPC*(TC2-TC1))
      REC=DIN*MFRC*4./(3.14*DIN*DIN)/MUC
      REW=DEQ*MFRW*4./(3.14*DEQ*DEQ)/MUW
      PRW=CPW*MUW/KW
      PRC=CPC*MUC/KC
      IF(REC.LT.2500.)GO TO 4
      X1=.092*MUC**.2/(32.17*ROC)
      GO TO 5
4     X1=32.*MUC/(32.17*ROC*REC*ROC*REC)
5     IF(REW.LT.2500.)GO TO 6
      X2=.092*MUW**.2/(32.17*ROW)
      GO TO 12
6     X2=32.*MUW/(32.17*ROW*ROW*ROW*REW)
12    IF(REC.LT.10000.) GO TO 32
      X3=.023*KC*(CPC*MUC/KC)**.4/MUC**.8
      GO TO 24
32    IF(REC.LT.2100.)GO TO 23
      X3=.166*KC*PRC**.33/(REC**.33*MUC**.8)
```

```
      GO TO 24
23    X3=1.86*KC*PRC**.5/(REC**.13*MUC**.8)
24    IF(REW.LT.10000.) GO TO 25
      X4=.023*KW*(CPW*MUW/KW)**.4/MUW**.8
      GO TO 7
25    IF(REW.LT.2100.)GO TO 26
      X4=.116*KW*PRW**.33/(REW**.13*MUW**.8)
      GO TO 7
26    X4=1.86*KW*PRW**.5/(REW**.13*MUW**.8)
7     X5=(PC1-PC2)/(PW1-PW2)
      X6=(MFRC*DCUT/MFRW/DIN)**.4*(X2*X5/X1)**.33
      X7=X3/X4*((X2*X5/X1)**.33)*((DCUT/DIN)**.2)*((MFRC/MFRW)**.2)
      X8=1./X3*(1.+(DIN*X7/DCUT))
      B=(TW1-TC2)/(TW2-TC1)
      DTLM=(TW1-TC2)-(TW2-TC1)/ALOG(B)
      X9=CPC*(TC2-TC1)/(4.*DTLM)
      GC=((PC1-PC2)/(X1*X8*X9))**.5
      GW=GC/X6
      D=MFRC*4./(GC*3.14*DIN*DIN)
      K=D
      N=FLOAT K
      DT=TW1-TW2
      DDT=DT/5.
      DTW1=TW1
      DO 100 J=1,5
      XTW1(J)=DTW1
      XTW2(J)=DTW1-DDT
      DTW1=XTW2(J)
100   CONTINUE
      DO 110 J=1,5
      TM=(XTW1(J)-XTW2(J))/2.+XTW2(J)
      IF(TM.LT.TW3)GO TO 101
      PART=(TM-TW3)/(TW1-TW3)
      XCPW(J)=PART*(DCPW1-DCPW3)+DCPW3
      XKW(J)=PART*(DKW1-DKW3)+DKW3
      XMUW(J)=PART*(DMUW1-DMUW3)+DMUW3
```

```
      XROW(J)=PART*(XROW1-XROW3)+XROW3
      GO TO 102
101   PART=(TM-TW2)/(TW3-TW2)
      XCPW(J)=PART*(DCPW3-DCPW2)+DCPW2
      XKW(J)=PART*(DKW3-DKW2)+DKW2
      XMUW(J)=PART*(DMUW3-DMUW2)+DMUW2
      XROW(J)=PART*(DROW3-DROW2)+DROW2
102   F=1.
110   CONTINUE
      DTC1=TC1
      DO 120 J=1,4
      XTC1(J)=DTC1
      DTC=WFRW*CP
      DTC=WFRW*XCPW(J)*(XTW1(J)-XTW2(J))/(WFRC*CPC)
      XTC2(J)=XTC1(J)+DTC
      DTC1=XTC2(J)
120   CONTINUE
      J=5
      XTC2(J)=TC2
      XTC1(J)=DTC1
      DO 130 J=1,5
      TM=(XTC2(J)-XTC1(J))/2.+XTC1(J)
      IF(TM.GT.TC3)GO TO 121
      PART=(TC3-TM)/(TC3-TC1)
      XCPC(J)=PART*(DCPC3-DCPC1)+DCPC1
      XKC(J)=PART*(DKC3-DKC1)+DKC1
      XMUC(J)=PART*(DMUC3-DMUC1)+DMUC1
      XROC(J)=PART*(DROC3-DROC1)+DROC1
      GO TO 122
121   PART=(TC2-TM)/(TC2-TC3)
      XCPC(J)=PART*(DCPC2-DCPC3)+DCPC3
      XKC(J)=PART*(DKC2-DKC3)+DKC3
      XMUC(J)=PART*(DMUC2-DMUC3)+DMUC3
      XROC(J)=PART*(DROC2-DROC3)+DROC3
122   F=1.
130   CONTINUE
```

```
      DO 140 J=1,5
      XPW1(J)=PW1+((TW1-XTW1(J))/(TW1-TW2))*(PW1-PW2)
      XPW2(J)=PW1+((TW1-XTW2(J))/(TW1-TW2))*(PW1-PW2)
      XPC1(J)=PC1+((TC1-XTC1(J))/(TC1-TC2))*(PC1-PC2)
      XPC2(J)=PC1+((TC1-XTC2(J))/(TC1-TC2))*(PC1-PC2)
140   CONTINUE
      DO 20 I=1,5
      XPRW(I)=XCPW(I)*XMUW(I)/XKW(I)
      XPRC(I)=XCPC(I)*XMUC(I)/XKC(I)
11    IF(REC.LT.2500.)GO TO 14
      XX1=.092*XMUC(I)**.2/(32.17*XROC(I))
      GO TO 15
14    XX1=32.*XMUC(I)/(32.17*XROC(I)**3.*REC)
15    IF(REW.LT.2500.)GO TO 16
      XX2=.092*XMUW(I)**.2/(32.17*XROW(I))
      GO TO 41
16    XX2=32.*XMUW(I)/(32.17*XROW(I)**3.*REW)
41    IF(REC.LT.10000.) GO TO 42
      XX3=.023*XKC(I)*(XCPC(I)*XMUC(I)/XKC(I))**.4/XMUC(I)**.8
      GO TO 44
42    IF(REC.LT.2100.) GO TO 43
      XX3=.116*XKC(I)*XPRC(I)**.33/(REC**.33*XMUC(I)**.8)
      GO TO 44
43    XX3=1.867XKC(I)*XPRC(I)**.5/(REC**.13*XMUC(I)**.8)
44    IF(REW.LT.10000.)GO TO 45
      XX4=.023*XKW(I)*(XCPW(I)*XMUW(I)/XKW(I))**.4/XMUW(I)**.8
      GO TO 17
45    IF(REW.LT.2100.) GO TO 46
      XX4=.116*XKW(I)*XPRW(I)**.33/(REW**.33*XMUW(I)**.8)
      GO TO 17
46    XX4=1.86*XKW(I)*XPRW(I)**.5/(REW**.13*XMUW
17    XX5=(XPC1(I)-XPC2(I))/(XPW1(I)-XPW2(I))
      XX6=(MFRC*DOUT/MFRW/DIN)**.4*((XX2*XX5/XX1)**.33)
      XX7=XX3/XX4*(XX2*XX5/XX1)**.33*(DOUT/DIN)**.2*(MFRC/MFRW)**.2
      XX8=1./XX3*(1.+DIN*XX7/DOUT)
      BB=(XTW1(I)-XTC2(I))/(XTW2(I)-XTC1(I))
```

```
      DTLM=(XTW1(I)-XTC2(I))-(XTW2(I)-XTC1(I))/ALOG(BB)
      XX9=XCFC(I)*(XTC2(I)-XTC1(I))/(4.*DTLM)
      XX10=GC
      XX11=N*DIN*DIN
      XX12=3.14*XX8*XX9*XX10**.2*XX11
      XLN(I)=XX9*XX8*GC**-.2*DIN**1.2
      XAREA(I)=XX12*DIN**.2
      XDPC(I)=XX1*GC**1.8*XLN(I) /(DIN**1.2)
       XDPW(I)=XX2*GW**1.8*XLN(I)/(DEQ**1.2)
 20   CONTINUE
      LENGTH=0.
      AREA=0.
      DPC=0.
      DPW=0.
      DO 21 I=1,5
      LENGTH=LENGTH+XLN(I)
      AREA=AREA+XAREA(I)
      DPC=DPC+XDPC(I)
      DPW=DPW+XDPW(I)
 21   CONTINUE
C     COST CALCULATIONS
      IF(DCUT.LE.0.042)CWV=3.0
      TLF=1.3-((LENGTH-3.)**.5*0.1)
      CWV=((12.0/5.0)*(DCUT-(.5/12.0)))+3.0
C     CW IS THE FABRICATION COSTS
      CW=CWV*N
C     TMF IS THE COST FACTOR FOR THE TYPE OF MATERIAL USED
C     CTB IS THE COST OF THE MATERIAL FOR THE TUBES AND THE SHELL
      CTB=((150.-((AREA-100.)**.5)/5.)/10.)*TLF*AREA*TMF
      COST=CW+CTB
      WRITE(5,2020)
      WRITE(5,1050)
      WRITE(5,2002) MFRW
      WRITE(5,2003) MFRC
      WRITE(5,2004) N
      WRITE(5,2005) DIN
```

```
      WRITE(5,2006) DOUT
      WRITE(5,2007) S
      WRITE(5,2008) LENGTH
      WRITE(5,2009) DPW
      WRITE(5,2010)DPC
      WRITE(5,2011) COST
60    CONTINUE
1000 FORMAT(5F12.4)
1001 FORMAT(8F10.4)
1002 FORMAT(7F10.4)
1003 FORMAT(I4)
1005 FORMAT(3F12.4)
1050 FORMAT(5X,'                         ')
2002 FORMAT(1X,'MASS FLOW RATE WORKING FLUID'5X,F12.4,1X,'LB/HR')
2003 FORMAT(1X,'MASS FLOW RATE COOLING FLUID'5X,F12.4,1X,'LB/HR')
2004 FORMAT(1X,'NUMBER OF TUBES'18X,F12.4)
2005 FORMAT(1X,'INNER DIAMETER OF TUBES'11X,F12.4,1X,'FEET')
2006 FORMAT(1X,'OUTER DIAMETER OF TUBES'11X,F12.4,1X,'FEET')
2007 FORMAT(1X,'SPACING OF TUBES'17X,F12.4,1X,'FEET')
2008 FORMAT(1X,'LENGTH OF TUBES'14X,F12.4,1X,'FEET')
2009 FORMAT(1X,'DROP IN PRESSURE WORKING FLUID'3X,F12.4,1X,'LB/SQ FT')
2010 FORMAT(1X,'DROP IN PRESSURE COOLING FLUID'3X,F12.4,1X,'LB/SQ FT')
2011 FORMAT(1X,'COMPARITIVE COSTS'17X,F12.2,3X,'DOLLARS')
2020 FORMAT(1H1)
      END
```

Appendix E. CROSS-FLOW HEATER HEAT EXCHANGER PROGRAM

E.1 Instruction for Operation and Inputs

Written in FORTRAN IV, this program is designed to be run on the Interdata computer of the Joint Civil and Mechanical Engineering Computer Facility at M.I.T. With modifications in the control, input and output cards, this program can be run on any computer that is designed to compile FORTRAN IV.

The first three cards of the input data are used for properties of the fluids. Ten cards are then needed to input the ten bt ten matrix F. The nest card contains a single number, M. This is the number of geometric arrangements that will be examined. The input cards that follow are in two card sets. The first of these contains an identification name, the second has the data on geometric arrangement, DIN, DOUT, S, N, and KMAT. With this input arrangement, M different geometric arrangements can be examined with the same fluid properties.

The input data is punched on the cards as follows:

(1) All values will be in the units of lb_m, hr., feet, Btu, and ^oF

(2) All values except M will be fixed point numbers and will have a decimal point. M is a floating point number.

(3) No commas are needed to separate the input data on a single card.

(4) The input values must be in the designated columns but do not need to be right or left justified.

(5) All fixed point numbers can have up to 4 (four) digits after the decimal point.

(6) The fluid properties are average values of these properties over the temperature range.

CARD	COLUMN					
	1--12	13--24	25--36	37--48	49--60	61--70
1	MFRW	TWH	TWC	THH	THC	PH1
2	PH2	PW1	PW2	CPW	CPH	MUW
3	MUH	KW	KH	ROH	ROW	

Cards 4 through 13 contain values of the matrix F

Theses values are punched 10 to a card with six columns per value

	1--4				
14	M				
	5--------18				
15	Identification name				
	1--12	13--24	25--36	37--48	49--60
16	DIN	DOUT	S	N	KMAT
17	Same as 15				
18	Same as 16				

These last two cards are repeated M times

The output will appear with the identification name at the top of a page and the following printed below it:

(cont. overleaf).

MASS FLOW RATE WORKING FLUID
MASS FLOW RATE HEATING FLUID
NUMBER OF TUBES
INNER DIAMETER OF TUBES
OUTER DIAMETER OF TUBES
SPACING OF TUBES
LENGTH OF TUBES
DROP IN PRESSURE WORKING FLUID
DROP IN PRESSURE HEATING FLUID
COMPARITIVE COSTS

For each identification name there will be a separate page

E.2 LIST OF VARIABLES

AREA	Heat transfer area in ft^2.
COST	Cost of exchanger in dollars
CPH	Specific heat at constant pressure of heating fluid in Btu/lb_m ^oF
CPW	Specific heat at constant pressure of working fluid in Btu/lb_m ^oF
CTB	Cost of materials for shell and tubes in dollars
CW	Fabrication costs in dollars
CWV	Fabrication costs per tube in dollars
DIN	Inner diameter of tubes in ft.
DOUT	Outer diameter of tubes in ft.
DLPH*	Frictional pressure drop in lb/ft^2.
EC	Heat exchanger effectiveness
F	Mean temperature difference ratio
GH	Mass flow velocity of heating fluid in $lb_m/hr.\ ft^2$.
GW	Mass flow velocity of working fluid in $lb_m/hr.\ ft^2$.
HH	Heat transfer coefficient of heating fluid in $Btu/hr.\ ft^2.\ ^oF$
HW	Heat transfer coefficient of working fluid in $Btu/hr.\ ft^2.\ ^oF$
KH	Thermal conductivity of heating fluid in $Btu/hr.\ ft^2.\ ^oF$
KW	Thermal conductivity of working fluid in $Btu/hr.\ ft.\ ^oF$
LENGTH	Length of heat exchanger in ft.
M	Number of input data groups
MFRH	Mass flow rate of heating fluid in lb_m/hr.
*DTLM	Log mean temperature difference in ^oF

LIST OF VARIABLES (E.2) (cont).

MFRW	Mass flow rate of working fluid in lb_m/hr
MUH	Viscosity of heating fluid in $lb_m/hr.\ ft.$
MUW	Viscosity of working fluid in $lb_m/hr.\ ft.$
N	Number of tubes
PH1	Inlet pressure of heating fluid in lb/ft^2.
PH2	Outlet pressure of heating fluid in lb/ft^2.
PW1	Inlet pressure of working fluid in lb/ft^2.
PH2	Outlet pressure of working fluid in lb/ft^2.
PRH	Prandtl number of heating fluid
PRW	Prandtl number of working fluid
Q	Heat transfer rate in Btu/hr.
REH	Reynolds number of heating fluid
REW	Reynolds number of working fluid
ROH	Density of heating fluid in lb_m/ft^3.
ROW	Density of working fluid in lb_m/ft^3.
S	Tube spacing
THC	Cooler temperature of heating fluid in ^oF
THH	Hotter temperature of heating fluid in ^oF
TWC	Cooler temperature of working fluid in ^oF
TWH	Hotter temperature of working fluid in ^oF
TLF	Tube length factor
TMF	Tube material factor
U	Overall heat transfer coefficient in Btu/hr $ft^2\ ^oF$
Z	Temperature coefficient

```
C     CROSS FLOW HEATER WITH WORKING FLUID IN SHELL AND MIXED
      DIMENSION F(10,10)
      REAL MFRW,MFRH,MUW,MUH,N,HW,HH,KW,KH,KMAT,LENGTH
      READ(8,1000)MFRW,TWH,TWC,THH,THC,PH1
      READ(8,1000)PH2,PW1,PW2,CPW,CPH,MUW
      READ(8,1002)MUH,KW,KH,RCH,RCW
      READ(8,1001)F
      READ(8,1003)M
      DO 40 L=1,M
      READ(8,1050)
      READ(8,1002)DIN,DOUT,S,N,KMAT
1     MFRH=MFRW*CPW*(TWH-TWC)/(CPH*(THH-THC))
      PRW=CPW*MUW/KW
      PRH=CPH*MUH/KH
      B=(THC-TWC)/(THH-TWC)
      DTLM=(THC-TWC)-(THH-TWH)/ALOG(B)
      Z=(THH-THC)/(TWH-TWC)
      EC=(TWH-TWC)/(THH-TWC)
      D=EC*10.
      I=D
      IF(Z.LT.2.) GO TO 2
      K=Z
      IF(K.EQ.J)J=1
      IF(K.EQ.3)J=2
      IF(K.EQ.2)J=3
      GO TO 8
2     IF (Z.GT.1.25)GO TO 3
      IF(Z.GT.0.9)GO TO 4
      IF(Z.GT.0.7)GO TO 5
      IF(Z.GT.0.5)GO TO 6
      IF(Z.GT.0.3)GO TO 7
      J=9
      GO TO 8
3     J=4
      GO TO 8
4     J=5
```

```
      GO TO 8
5     J=6
      GO TO 8
6     J=7
      GO TO 8
7     J=8
8     Q=MFRW*CPW*(TWH-TWC)
      GH=MFRH*4./(3.14*DIN*DIN*N)
      GW=MFRW/((S-DCUT)**2.*N)
      REW=(S-DCUT)*GW/MUW
      REH=GH*DIN/MUH
      IF(REW.LT.10000.)GO TO 9
      HW=.023*KW*(CPW*MUW/KW)**.4*GW**.8/MUW**.8/(S-DCUT)**.2
      GO TO 10
9     IF(REW.LT.2100.) GO TO 19
      HW=.116*KW*PRW**.33*GW**.67/(S-DCUT)**.33/MUW**.67
      GO TO 10
19    HW=1.86*KW*PRW**.5*GW**.5/(S-DCUT)**.5/MUW**.5
10    IF(REH.LT.10000.)GO TO 11
      HH=.023*KH*(CPH*MUH/KH)**.4*GH**.8/MUH**.8/DIN**.2
      GO TO 12
11    IF(REH.LT.2100.) GO TO 22
      HH=.116*KH*PRH**.33*GH**.67/MUH**.67/DIN**.33
      GO TO 12
22    HH=1.86*KH*PRH**.5*GH**.5/DIN**.5/MUH**.5
12    UU=(DOUT-DIN)/KMAT+1./HH+1./HW
      U=1./UU
      AREA=Q/(U*F(I,J)*DTLM)
      LENGTH=AREA/(3.14*DIN*N)
      IF(REH.LT.2500.)GO TO 15
      DLPH=(.4*.046/REH**.2)*(MFRH*4./(3.14*DIN*DIN))**.2*(5./(32.17*
     1RCH))*LENGTH
      GO TO 20
15    DLPH=128.*MUH*GH*LENGTH/(32.17*ROH*DIN*DIN)
20    PH=PH1-PH2
      IF(DLPH.LE.PH)GO TO 30
```

```
      DD=DOUT/DIN
      DS=S/DOUT
      DIN=DIN+.01
      DOUT=DD*DIN
      S=DS*DOUT
      GO TO 1
 30   CONTINUE
C     COST CALCULATIONS
      IF(DOUT.LE.0.042)CWV=3.0
      TLF=1.3-((LENGTH-8.)**.5*0.1)
      CWV=((12.0/5.0)*(DOUT-(.5/12.0)))+3.0
C     CW IS THE FABRICATION COSTS
      CW=CWV*N
C     TMF IS THE COST FACTOR FOR THE TYPE OF MATERIAL USED
C     CTB IS THE COST OF THE MATERIAL FOR THE TUBES AND THE SHELL
      CTB=((150.-((AREA-100.)**.5)/5.)/10.)*TLF*AREA*TMF
      COST=CW+CTB
      WRITE(5,2020)
      WRITE(5,1050)
      WRITE(5,2002) MFRW
      WRITE(5,2003) MFRH
      WRITE(5,2004) N
      WRITE(5,2005) DIN
      WRITE(5,2006) DOUT
      WRITE(5,2007) S
      WRITE(5,2009) LENGTH
      WRITE(5,2010) DLPH
      WRITE(5,2011) COST
 40   CONTINUE
 1000 FORMAT(6F12.4)
 1001 FORMAT(10F6.2)
 1002 FORMAT(5F12.4)
 1003 FORMAT(I4)
 1050 FORMAT(5X,'                    ')
 2002 FORMAT(1X,'MASS FLOW RATE WORKING FLUID'5X,F12.4,1X,'LB/HR')
 2003 FORMAT(1X,'MASS FLOW RATE HEATING FLUID'5X,F12.4,1X,'LB/HR')
```

```
2004 FORMAT(1X,'NUMBER OF TUBES'18X,F12.4)
2005 FORMAT(1X,'INNER DIAMETER OF TUBES'11X,F12.4,1X,'FEET')
2006 FORMAT(1X,'OUTER DIAMETER OF TUBES'11X,F12.4,1X,'FEET')
2007 FORMAT(1X,'SPACING OF TUBES'17X,F12.4,1X,'FEET')
2008 FORMAT(1X,'LENGTH OF TUBES'14X,F12.4,1X,'FEET')
2009 FORMAT(1X,'DROP IN PRESSURE WORKING FLUID'3X,F12.4,1X,'LB/SQ FT')
2010 FORMAT(1X,'DROP IN PRESSURE HEATING FLUID'3X,F12.4,1X,'LB/SQ FT')
2011 FORMAT(1X,'COMPARITIVE COSTS'17X,F12.2,3X,'DOLLARS')
2020 FORMAT(1H1)
     END
```

Appendix F. CROSS-FLOW COOLER HEAT EXCHANGER PROGRAM

F.1 Instructions for Operation and Inputs

Written in FORTRAN IV, this program is designed to be run on the Interdata computer of the Joint Civil and Mechanical Engineering Computer Facility at M.I.T. With modifications in the control, input and output cards, this program can be run on any computer that is designed to compile FORTRAN IV.

The first three cards of the input deck contain properties for the fluids. The next ten cards are used to input the ten by ten matrix F. A single number, M, is placed on the fourteenth card. This is the number of geometric arrangements that will be examined. The input cards that follow are in two card sets. The first of these two cards contains an identification name. The second contains the data on the geometrical arrangement of the exchanger, DOUT, DIN, S, N, and KMAT. With this input arrangement, M different geometric arrangements can be examined using the same fluid properties.

The input data is punched on cards as follows:

(1) All values will be in units of lb_m, hr., feet, Btu, and ^oF

(2) All values except M will have fixed point numbers and will have a decimal point. M is a floating point number

(3) No commas are needed to separate the input data on a single card

(4) The input values must be in the designated column but need not be right or left justified

(5) All fixed point numbers can have up to 4 (four) digits after the decimal point.

(6) The fluid properties are average values averaged over the temperature range

CARD	COLUMN					
	1--12	13--24	25--36	37--48	49--60	61--72
1	MFRW	TCH	TCC	TWH	TWC	PC1
2	PC2	PW1	PW2	CPW	CPC	MUC
3	MUW	KW	KC	ROW	ROC	

Cards 4 through 13 contain values of the matrix F

These values are punched 10 to a card with 6 columns per value

	1---4				
14	M				
	5--------18				
15	Identification name				
	1--12	13--24	25--36	37--48	49--60
16	DIN	DOUT	S	N	KMAT
17	Same as 15				
18	Same as 16				

These last two cards are repeated M times

The output will appear with the identification name at the top of a page and the following printed below it:

MASS FLOW RATE WORKING FLUID
MASS FLOW RATE COOLING FLUID
NUMBER OF TUBES
INNER DIAMETER OF TUBES

(cont. overleaf)

OUTER DIAMETER OF TUBES
SPACING OF TUBES
LENGTH OF TUBES
DROP IN PRESSURE WORKING FLUID
DROP IN PRESSURE COOLING FLUID
COMPARITIVE COSTS

For each identification name there will be a separate page.

F.2 LIST OF VARIABLES

AREA	Heat transfer surface area in ft^2.
COST	Cost of exchanger in dollars
CPC	Specific heat at constant pressure of cooling fluid in Btu/lb_m ^{o}F
CPW	Specific heat at constant pressure of working fluid in Btu/lb_m ^{o}F
CTB	Cost of materials for shell and tubes in dollars
CW	Fabrication costs in dollars
CWV	Fabrication costs per tube in dollars
DIN	Inner diameter of tubes in ft.
DOUT	Outer diameter of tubes in ft.
DLPC	Frictional pressure drop of cooling fluid in lb/ft^2.
DTLM	Log mean temperature difference in ^{o}F
EC	Heat exchanger effectiveness
F	Mean temperature difference factor
GC	Mass flow velocity of cooling fluid in lb_m/hr. ft^2.
GW	Mass flow velocity of working fluid in lb_m/hr. ft^2.
HC	Heat transfer coefficient of cooling fluid in Btu/hr. ft^2. ^{o}F
HW	Heat transfer coefficient of working fluid in Btu/hr. ft^2. ^{o}F
KC	Thermal conductivity of cooling fluid in Btu/hr. ft. ^{o}F
KW	Thermal conductivity of working fluid in Btu/hr. ft. ^{o}F
LENGTH	Length of heat exchanger in ft.

F.2 LIST OF VARIABLES (cont).

M	Number of input data groups
MFRC	Mass flow rate of cooling fluid in lb_m/hr.
MFRW	Mass flow rate of working fluid in lb_m/hr.
MUC	Viscosity of cooling fluid in lb_m/ft.
MUW	Viscosity of working fluid in lb_m/ft.
N	Number of tubes
PC1	Inlet pressure of cooling fluid in lb/ft^2.
PC2	Outlet pressure of cooling fluid in lb/ft^2.
PW1	Inlet pressure of working fluid in lb/ft^2.
PW2	Outlet pressure of working fluid in lb/ft^2.
PRC	Prandtl number of cooling fluid
PRW	Prandtl number of working fluid
Q	Heat transfer rate in Btu/hr.
REC	Reynolds number of cooling fluid
REW	Reynolds number of working fluid
ROC	Density of cooling fluid in lb_m/ft^3.
ROW	Density of working fluid in lb_m/ft^3.
S	Tube spacing
TCC	Cooler temperature of cooling fluid in ^oF
TCH	Hotter temperature of cooling fluid in ^oF
TWC	Cooler temperature of working fluid in ^oF
TWH	Hotter temperature of working fluid in ^oF
TLF	Tube length factor

F.2 LIST OF VARIABLES (cont).

TMF	Tube material factor
U	Overall heat transfer coefficient in Btu/hr. ft^2. ^{o}F
Z	Temperature coefficient

```
C       CROSS FLOW COOLER WITH WORKING FLUID IN SHELL MIXED
        DIMENSION F(10,10)
        REAL MFRC,MFRW,HW,KW,HC,KC,MUW,MUC,KMAT,LENGTH,N
        READ(8,1000)MFRW,TCH,TCC,TWH,TWC,PC1
        READ(8,1000)PC2,PW1,PW2,CPW,CPC,MUC
        READ(8,1002)MUW,KW,KC,RCW,RCC
        READ(8,1001)F
        READ(8,1003)M
        DO 40 L=1,M
        READ(8,1050)
        READ(8,1002)DIN,DOUT,S,N,KMAT
  1     MFRC=MFRW*CPW*(TWH-TWC)/(CPC*(TCH-TCC))
        PRW=CPW*MUW/KW
        PRC=CPC*MUC/KC
        B=(TWC-TCC)/(TWH-TCC)
        DTLM=(TWC-TCC)-(TWH-TCH)/ALOG(B)
        Z=(TWH-TWC)/(TCH-TCC)
        EC=(TCH-TCC)/(TWH-TCC)
        D=EC*10.
        I=D
        IF(Z.LT.2.) GO TO 2
        K=Z
        IF(K.EQ.J)J=1
        IF(K.EQ.3)J=2
        IF(K.EQ.2)J=3
        GO TO 8
  2     IF (Z.GT.1.25)GO TO 3
        IF(Z.GT.0.9)GO TO 4
        IF(Z.GT.0.7)GO TO 5
        IF(Z.GT.0.5)GO TO 6
        IF(Z.GT.0.3)GO TO 7
        J=9
        GO TO 8
  3     J=4
        GO TO 8
  4     J=5
```

```
      GO TO 8
5     J=6
      GO TO 8
6     J=7
      GO TO 8
7     J=8
8     Q=MFRW*CPW*(TWH-TWC)
      GW=MFRW/((S-DOUT)**2.*N)
      GC=MFRC*4./(3.14*DIN*DIN*N)
      REC=DIN*GC/MUC
      REW=(S-DOUT)*GW/MUW
      IF(REW.LT.10000.)GO TO 9
      HW=.023*KW*(CPW*MUW/KW)**.4*GW**.8/MUW**.8/(S-DOUT)**.2
      GO TO 10
9     IF(REW.LT.2100.) GO TO 19
      HW=.116*KW*PRW**.33*GW**.67/(S-DOUT)**.33/MUW**.67
      GO TO 10
19    HW=1.86*KW*PRW**.5*GW**.5/(S-DOUT)**.5/MUW**.5
10    IF(REC.LT.10000.)GO TO 11
      HC=.023*KC*(CPC*MUC/KC)**.4*GC**.8/MUC**.8/DIN**.2
      GO TO 12
11    IF(REC.LT.2100.) GO TO 22
      HC=.116*KC*PRC**.33*GC**.67/MUC**.67/DIN**.33
      GO TO 12
22    HC=1.86*KC*PRC**.5*GC**.5/DIN**.5/MUC**.5
12    UU=(DOUT-DIN)/KMAT+1./HC+1./HW
      U=1./UU
      AREA=Q/(U*F(I,J)*DTLM)
      LENGTH=AREA/(3.14*DIN*N)
      IF(REC.LT.2500.)GO TO 15
      DLPC=(.4*.046/REC**.2)*(MFRC*4./(3.14*DIN*DIN))**2.*(5./(32.17*ROC
     1))*LENGTH
      GO TO 20
15    DLPC=128.*MUC*GC*LENGTH/(32.17*ROC*DIN*DIN)
20    PC=PC1-PC2
      IF(DLPC.LE.PC)GO TO 30
```

```
      DL=DOUT/DIN
      DS=S/DOUT
      DIN=DIN+.01
      DOUT=DL*DIN
      S=DS*DOUT
      GO TO 1
 30   CONTINUE
C     COST CALCULATIONS
      IF(DOUT.LE.0.042)CWV=3.0
      TLF=1.3-((LENGTH-8.)**.5*0.1)
      CWV=((12.0/5.0)*(DOUT-(.5/12.0)))+3.0
C     CW IS THE FABRICATION COSTS
      CW=CWV*N
C     TMF IS THE COST FACTOR FOR THE TYPE OF MATERIAL USED
      TMF=2.5
C     CTB IS THE COST OF THE MATERIAL FOR THE TUBES AND THE SHELL
      CTB=((150.-((AREA-100.)**.5)/5.)/10.)*TLF*AREA*TMF
      COST=CW+CTB
      WRITE(5,2020)
      WRITE(5,1050)
      WRITE(5,2002) MFRW
      WRITE(5,2003) MFRC
      WRITE(5,2004) N
      WRITE(5,2005) DIN
      WRITE(5,2006) DOUT
      WRITE(5,2007) S
      WRITE(5,2008) LENGTH
      WRITE(5,2010) DLPC
      WRITE(5,2011) COST
 40   CONTINUE
1000  FORMAT(6F12.4)
1001  FORMAT(10F6.2)
1002  FORMAT(5F12.4)
1003  FORMAT(I4)
1050  FORMAT(5X,'                        ')
2002  FORMAT(1X,'MASS FLOW RATE WORKING FLUID'5X,F12.4,1X,'LB/HR')
```

```
2003 FORMAT(1X,'MASS FLOW RATE COOLING FLUID'5X,F12.4,1X,'LB/HR')
2004 FORMAT(1X,'NUMBER OF TUBES'18X,F12.4)
2005 FORMAT(1X,'INNER DIAMETER OF TUBES'11X,F12.4,1X,'FEET')
2006 FORMAT(1X,'OUTER DIAMETER OF TUBES'11X,F12.4,1X,'FEET')
2007 FORMAT(1X,'SPACING OF TUBES'17X,F12.4,1X,'FEET')
2008 FORMAT(1X,'LENGTH OF TUBES'14X,F12.4,1X,'FEET')
2009 FORMAT(1X,'DROP IN PRESSURE WORKING FLUID'3X,F12.4,1X,'LB/SQ FT')
2010 FORMAT(1X,'DROP IN PRESSURE COOLING FLUID'3X,F12.4,1X,'LB/SQ FT')
2011 FORMAT(1X,'COMPARITIVE COSTS'17X,F12.2,3X,'DOLLARS')
2020 FORMAT(1H1)
     END
```

Appendix G - EXAMINATION OF THE ZENER CYCLE

G.1 Examination

To examine the usefulness of the computer progrmas developed, the thermal cycle proposed by Prof. Clarence Zener(1) was examined. The program for designing condensers and the program for evaporator design were used.

In his proposal, Zener suggests the use of ammonia as the working fluid. The condensor and the evaporator for this cycle were designed using ammonia and freon-21 as working fluids. Becaose of its high latent heat of vaporization about 540 Btu/lb_m, ammonia requires a much larger heat exchanger for evaporation than does freon-21 with a latent heat of vaporization of 102 Btu/lb_m. This difference in size can be seen in figures G-1, G-2 and G-3.

The properties of Zener's proposed cycle, as given In Ref. 1, were used as inputs to the computer programs. The allowable pressure drop over the length of the condenser and evaporator was used as a constraint on the size of the exchangers. An inner diameter of ¼ inch was used as a starting diameter. If this diameter caused a frictional pressure drop greater than the allowed pressure drop, the diameter of the tubes was increased and the heat exchanger redesigned.

In the condenser design, two values of the coefficient of heat transfer for condensation were used. The program was first run with the coefficient being calculated as discussed in section

2.3. The design was then revised using a coefficient of 800 Btu/hr-ft^2 - $^\circ$F. This is the value thatGregorig(33) obtained using fluted tubes. This is further discussed in section 2.5. Zener in his proposal suggested the use of this type of surface to increase the heat transfer coefficient.

The two programs were run with the variation mentioned above and with different geometries. The size of the heat exchangers required for these different arrangements were then calculated.

G.2 Results

The condensor was sized using different numbers of tubes. The resultant required length of heat exchanger for each different number of tubes are plotted in Figure G-1. As there is a significant improvement in reducing the size of the required condensor with the use of Gregorig surfaces, it is that size that is plotted. Plotted on Figure G-1 are the results for both ammonia and freon-21. The minimum allowable diameter of the tubes for each given number of tubes is also plotted. Figure G-2.

The evaporator was sized in the same manner as the condenser. Zener suggests the use of surface enhancement for the evaporator also. However, with the high mass flow rate surface, enhancement appears to be of little advantage. The required length for the evaporator with the use of both ammonia and freon-21 is plotted in Figure G-3.

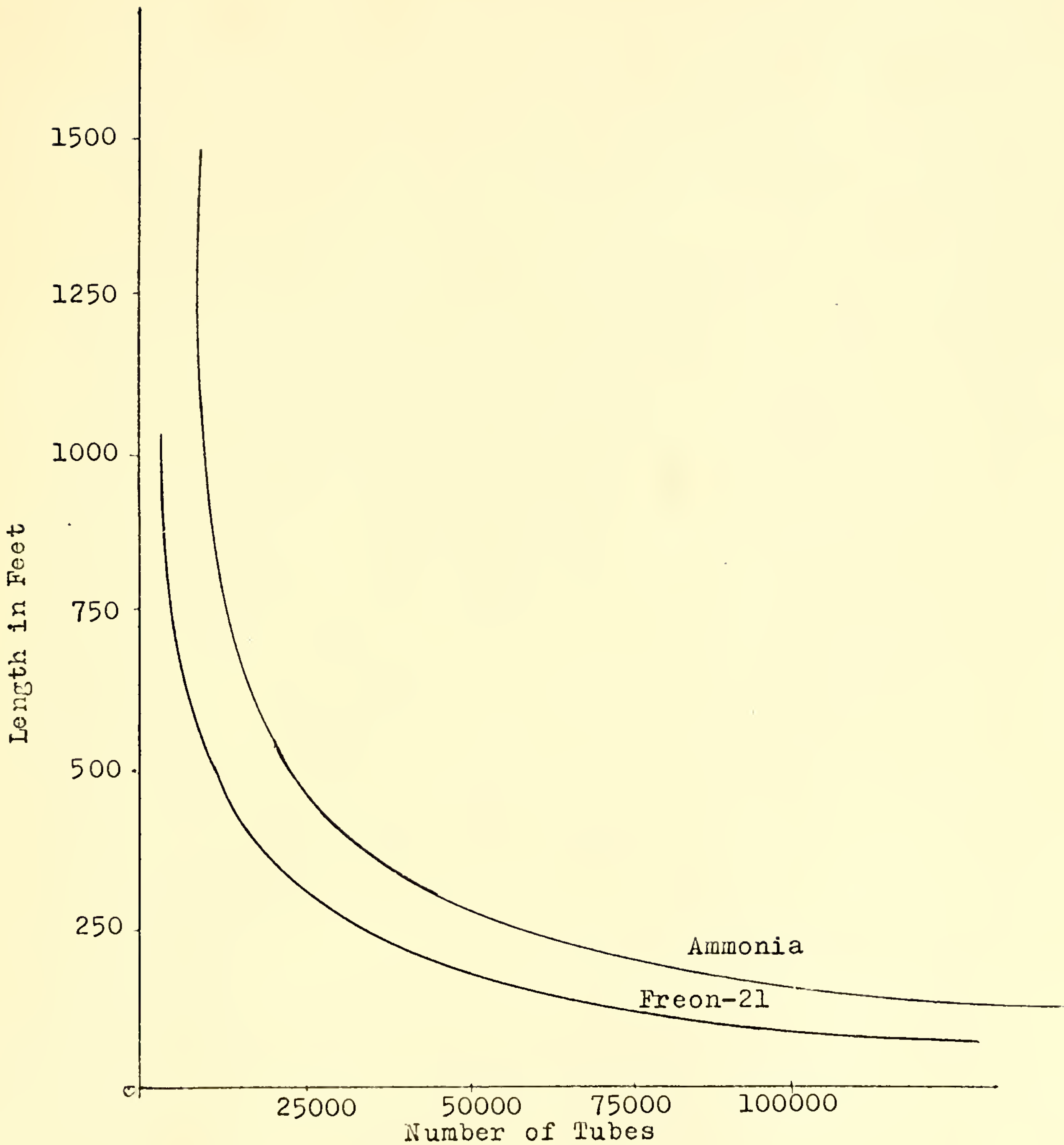

Figure G-1

Zener Cycle Condenser Number of Tubes vers Length

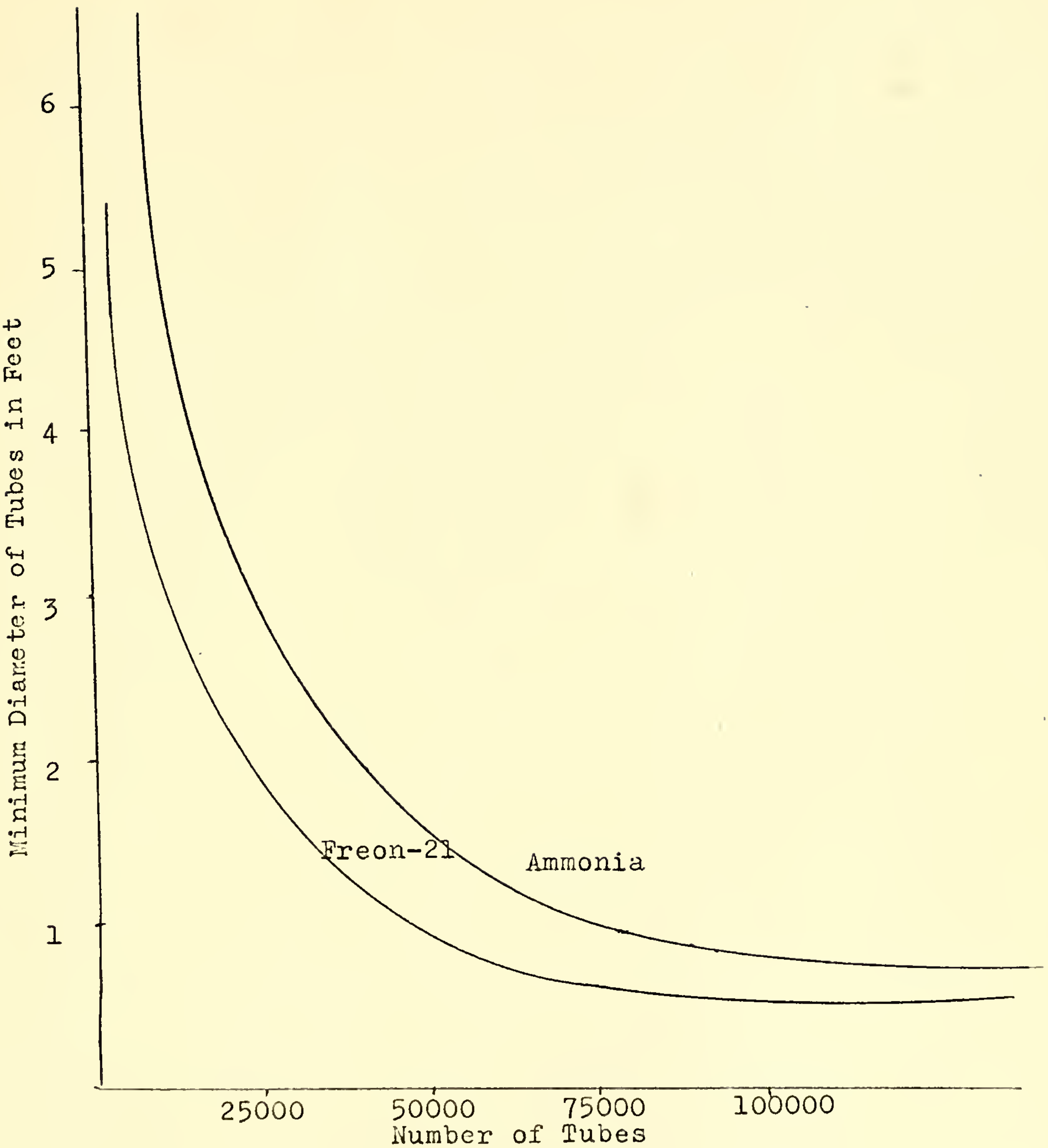

Figure G-2

Zener Cycle Condenser Number of Tubes vers
Tube Diameter

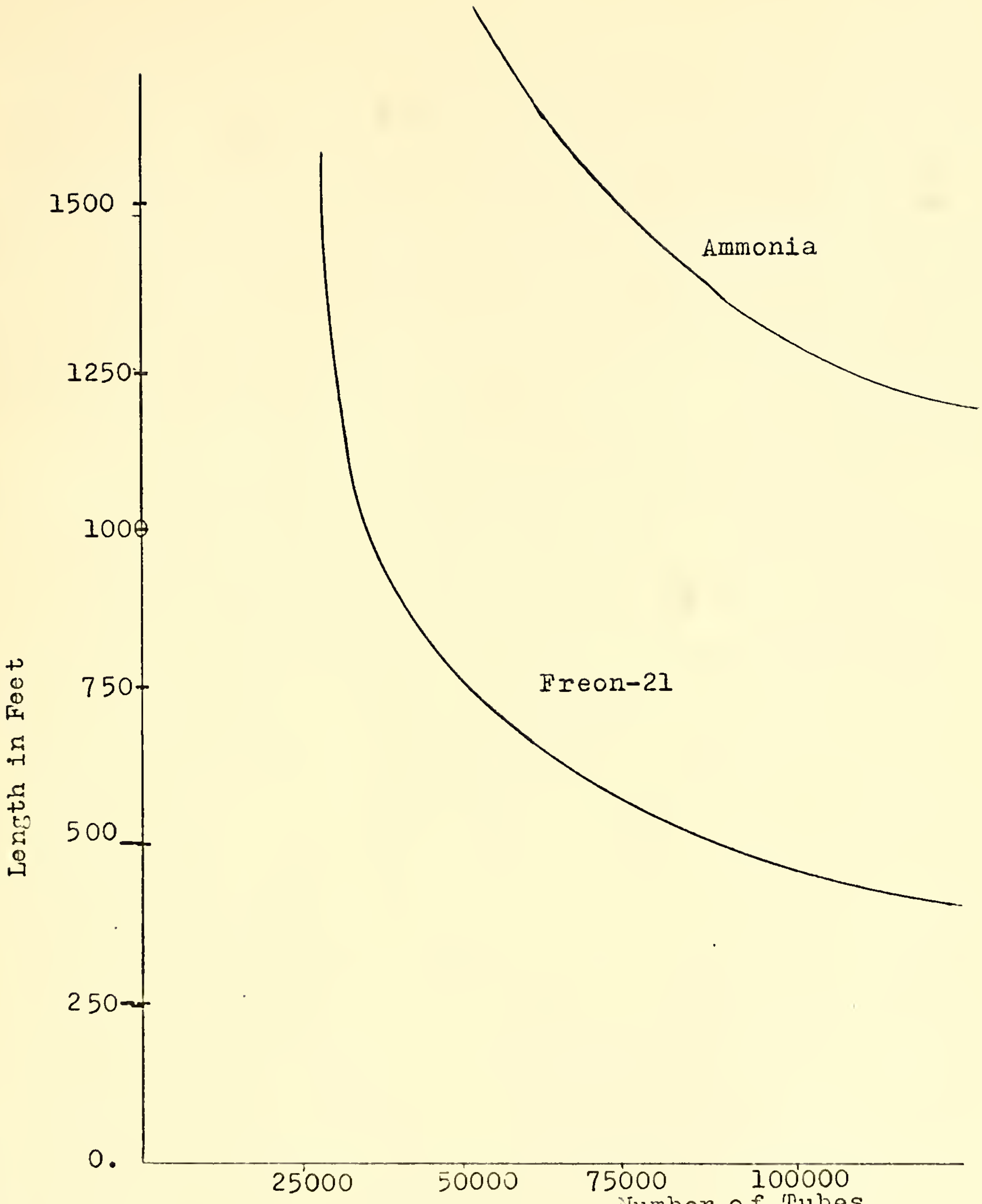

Figure G-3

Zener Cycle Evaporator Number of Tubes vers Length

G.3 Conclusions of the Zener Proposal

The use of ammonia as a working fluid in this cycle appears to be totally unrealistic. The minimum size requirement for the evaporator would amount to over thirty billion cubic feet with a length of over four thousand feet. The condenser using fluted tubes would be a great deal smaller, but still is over one billion cubic feet in total volume.

Freon-21 is a much better working fluid for these heat exchangers. The reduction in latent heat required to vaporize the working fluid reduces the size requirement of the evaporator. Still the total volume required for the evaporator is two billion cubic feet. The condenser for a freon-21 system would be over three hundred million cubic feet in volume.

The sizes of the above mentioned heat exchangers would be for production of 100 Mega-watts of electrical power. This is not an extremely large power plant by today's standards in terms of power output.

From a thermodynamic point of view, this cycle is a feasible one. However, when the size of the required heat exchangers is taken into account, the cycle quickly appears to be unrealistic. With a temperature difference of only two degrees Centigrade, in the evaporator, the mass flow rate is so great that the cycle is unusable.

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Thesis
J224 Jackson
Heat exchanger design
for thermal cycle feasi-
bility evaluation.